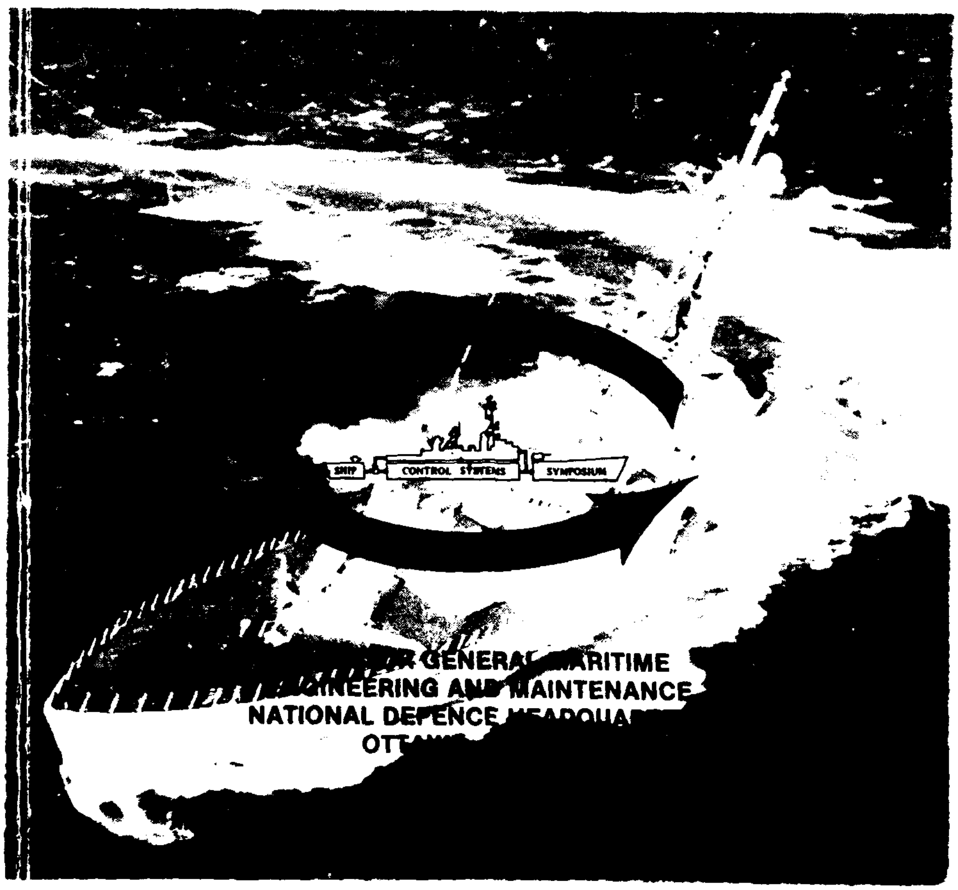
PROCEEDINGS

SIXTH
SHIP CONTROL SYSTEMS
SYMPOSIUM

26 - 30 OCTOBER 1981

VOLUME 4



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FOREWORD

The Director General Maritime Engineering and Maintenance (DGMEM) is pleased to present the Proceedings of the Sixth Ship Control System Symposium held at the Chateau Laurier/National Conference Centre complex in Ottawa, Canada, 26-30 October 1981. This is the sixth in a series of symposia on ship control systems. The First Ship Control Systems Symposium was convened in 1966.

The technical papers presented at the Symposium and published in these proceedings cover the entire spectrum of ship control systems and provide an insight into technological developments which are continuously offering the ship control system designer new options in addressing the complex man/machine operation. The microprocessor and its apparently unlimited development potential in future digital, distributed control systems appears ready to reshape the conventional concepts now so familiar in control system designs. There are many concerns that the advantages of the new technologies will be negated by the inability of training systems to graduate technicians who can adequately cope with these new systems.

The response to "Call For Papers" was outstanding and the papers selection committee constrained by the time available for presentations, was hard pressed to make their final selections from the many fine abstracts submitted. The final papers represent a unique international flavour which includes authors from every facet of the ship control system community. The final program is a balance of both theoretical and practical control system papers.

These Proceedings constitute the major record of the Sixth Ship Control Systems Symposium. The contents indicate the success of the Symposium and provide some insight into the effort that was required to ensure this success. The Symposium organizing committee, advisory groups, publications branch, authors, session chairmen, international coordinators, clerical and administrative personnel, and management all provided positive and cooperative support to the many tasks that had to be performed in organizing and presenting the Symposium.

This Symposium has continued to explore and present a number of specific aspects of ship control systems and undoubtly the next symposium will include new concepts and ideas which were unavailable for this Symposium. As in the past, we hope these Proceedings become a source document on ship control along with the previous proceedings. It is our hope that the Symposium has provided stimulation to those who will continue to advance this technical field.

Bruce H. Baxter
General Chairman

Philip V. Penny
Technical Chairman

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PROPULSION AND AUXILIARIES SECONDARY SURVEILLANCE SYSTEM (PASS) - A COMPREHENSIVE SURVEILLANCE SYSTEM FOR FUTURE GENERATION ROYAL NAVY WARSHIPS

by Lt. Cdr. B. Semke, RN, and
A. W. MacDonald, Y-ARD LTD.

INTRODUCTION

The following paper describes the development of a computer-based machinery surveillance system for future warships of the Royal Navy, in terms of the UK Ministry of Defence work in evolving the system requirements, and the implementation of these requirements by MOD's consultants, Y-ARD LTD., in the early stages of development.

BACKGROUND

In the 1950s the RN began to design ships with Machinery Control Rooms (MCRs) which provided for the centralised control and surveillance of steam propulsion machinery and related auxiliary systems. This trend towards unmanned machinery spaces provided the necessary protection from the hazards of nuclear, biological and chemical contamination as well as enabling reductions in the watchkeeping team. Conditions on watch improved but, apart from the extra remote controls, the watchkeepers saw little change in the type of surveillance offered; this being primarily by analogue gauges. Some further increases in the level of automation of these steam plants, using more pneumatic controls, were developed, but by then new propulsion systems had arrived.

The introduction of gas turbines into RN warship designs, in the late 1960s, together with controllable pitch propellers (CPP), brought a further increase in the level of both automatic and co-ordinated controls. These control systems were based on frequency analogue electronics and, after early problems, have proved to be very reliable. Development of the centralised control and surveillance philosophy also saw the introduction of the Ship Control Centre (SCC) where control and surveillance of most of the ship's machinery and systems was carried out by an even smaller watchkeeping team. Figure 1 shows how far we have come in reducing our watchkeeping teams.

Surveillance was now moving slowly away from analogue gauges and being enhanced by a rationalised alarms/warnings system, Decca Isis, which also provided a limited logging facility. The importance of "systems understanding" was being perceived and yet these techniques of surveillance in the SCC were not making the operating task as easy for the watchkeepers as perhaps it could have been. Improvements in the effectiveness of watchkeepers, however, and even further reductions in their numbers and skill level, demanded rather more than just more automation. The result was that MOD instituted a detailed study into the whole question of future technologies and the relationship between automation and manning. The Machinery Control and Surveillance (MC and S) research programme (Reference 1) began in 1975 and now, some 6 years later, MOD's developments are largely based on the recommendations of these studies.

THE NEW DESIGN CONCEPT

During the MC and S research programme the concept of a surveillance system separate from the machinery console was recommended for surface warships. This concept was already being developed for future submarine control rooms and the two developments now maintain a close liaison. This was made possible by the fact that MOD's long-standing consultants, Y-ARD LTD. of Glasgow, Scotland, were deeply involved from the outset in both contracts, and have been responsible for thc implementation of both prototype systems.

The concept first recognised that there was a considerable proportion of the SCC console devoted to surveillance which the operator had little need to be concerned with for the majority of his time on watch. It had the effect of needlessly complicating the task of the operator and thereby contributed to higher skill levels of personnel being required. In addition there was a growing requirement for more sophisticated surveillance information, particularly in the nuclear field, but also, with the growth of condition and performance monitoring techniques, on other plants and systems. Finally there was a continuing need to record data, which inevitably absorbed much time and effort on the watchkeeper's part.

The solution, in both surface and submarinc systems, was seen to be a computer-based surveillance system using visual display units (VDUs) as the operator interface. In both cases this secondary system was to be a complement to the primary system, which still included the controls, the alarms/warnings indications and some console surveillance. The actual implementation obviously diverged although there remains a considerable degree of commonality between them, especially in the software.

The surface ship system, known as the Propulsion and Auxiliaries Secondary Surveillance System (PASS), assumes two fundamental definitions :

(a) Primary Surveillance includes those parameters which are essential for the safe control and management of plant and systems at any of the operating positions. This, therefore, includes all alarms/warnings, indications and gauges mounted on the console, with the means for effecting control (e.g. levers, pushbuttons, etc.). These facilities are included in the Digital Propulsion Control (Demonstrator) system. (Reference 2).

(b) Secondary Surveillance provides for the monitoring of all parameters necessary for use by either operator or maintainer. It will provide more detailed information on plant and system state, including derived parameters, displaying them via the VDUs, in a more meaningful and more easily-assimilated form than in previous systems.

By dividing the total surveillance requirements in this way it will be possible to assist the operator in the safe monitoring of systems and the maintainer in his fault diagnosis and maintenance planning. The additional benefits of automated data logging will accrue to both the ship and the shore authorities who process the information.

Development Programme

The programme for development of this system is divided into several stages in order to provide good control of progress and cost. It has also enabled the development to remain largely independent of long term ship programmes, which is particularly useful when very tight financial controls are being applied and these long term plans are very fluid.

Stage 1 is basically a conceptual prototype stage, in which the system structure, in both hardware and software terms, is established, general ideas are proven, and the display formats for main propulsion machinery are evaluated. The basis for the application software is a fictional Reference Ship which is typical of the warships currently under consideration but which is not subject to regular changes during the early ship design stages.

Stage 2 will consist of further detailed definition of surveillance requirements, particularly for auxiliary machinery, the development of applications software to satisfy these requirements, and the definition of the target hardware required to satisfy both performance and environmental constraints of the shipboard application. The timing of this Stage, in relation to the ship programme, is such that the level of ship definition is sufficient to begin software definition but is not final. This will allow some interaction and discussion between ship requirements and PASS system facilities.

It is envisaged that a major subcontractor will be involved in the system development during this stage, and that this subcontractor will later become the main contractor responsible for the production, installation and support of shipboard systems.

Stage 3 and beyond will provide for the transfer of the core (executive) software to the target hardware, the production of the application software and the total system integration and evaluation ashore prior to first of class installation. This pre-production system will remain ashore to provide a facility for production and evaluation of modifications and to support and control the issue of all software in service.

The Prototype System - Concepts

It was recognised, during the preliminary work leading up to the decision to develop PASS, that the lifetime being projected for the system would be likely to span a number of changes in computer architecture and electronic hardware technology. Consequently, the decision was taken, fundamental to the system structure, that the software should be readily transportable, i.e. as independent of the hardware as possible. The detailed requirements of the operator and maintainer were to be implemented in application software, together with the necessary core software to operate the system, access and calculate data and interface to the various peripherals. Software standards and controls were to be of paramount importance from the outset because of the need to support and modify the software for many years to come, and the use of high level language (CORAL 66) and modular programme construction (MASCOT) was specified, (see Reference 3).

This approach paralleled that taken for the Digital Propulsion Control system, and as a result of this the same structure was adopted for the executive software of the PASS system as for the primary control system. The similarity of the interprocessor communication structures in turn allowed the use of Plant Control Units (part of the primary control system) as data sources for secondary surveillance of the propulsion plant. It should be borne in mind that secondary surveillance includes most of the parameters required for primary control and surveillance, presenting them in different formats to suit the different user requirements. It follows, therefore, that digital control systems for other items of ship machinery could readily interface to PASS, provided that compatibility of software structures and data communications is maintained.

An immediate effect of this approach to the software structure was to allow the selection of commercial grade, readily available, hardware for implementation of the prototype system, with minimum effort involved at a later stage in transfer to different hardware. The equipment for this stage of the project was selected from the Digital Equipment Corporation (DEC) PDP11 series, since this equipment is widely used and understood, and entailed relatively low cost and low development risk for system evaluation.

One of the major objectives in the development of PASS is to maximise the commonality of hardware with other ship systems, to achieve the benefits of increased production volume, minimised on-board spares holding, and reduced diversity of skills for on-board maintenance. To this end units required for data acquisition from ship plant not equipped with compatible digital control systems will be based on the hardware of the Demonstrator Plant Control Units of the digital propulsion control system.

Since the prototype system was based on the Reference Ship mentioned earlier, a further contribution to cost minimisation was achieved by implementing system facilities with a representative selection of parameters rather than the full complement to be expected in a real ship application. This allowed reduction of equipment costs and software production costs. However, if this approach to the early stages of system development is adopted, it is essential to incorporate a test phase which will exercise the system to the fullest extent in terms of sizing and speed so that possible areas of difficulty in a full scale system may be identified and eliminated.

Features

The particular facilities to be made available to the operator and maintainer in the prototype system were :

(a) Parameter pages displaying information on each plant and system via the VDU e.g. all pressures and temperatures in the lub oil system with a simple system schematic.

(b) History pages on the VDU giving individual parameter information over a specified period e.g. parameter readings over the last hour of operation.

(c) Logging facility, operating both automatically and on demand, outputting via the printer, the VDU or the portable data storage.

(d) Maintainer-assist pages providing any calculated data, including hours run, number of starts, trend graphs and performance calculations displayed on VDU.

The basic page type is the parameter page (Figure 2) which displays the current values, as sampled at 1 second intervals, of the main parameters associated with a particular item, or section of an item of ship machinery. Analogue bar-graph elements are used to facilitate the rapid appreciation of the general trend of plant behaviour, with a digital display of the current value under each bar-graph. A simplified schematic of the plant is used to indicate the positions of, and relationships amongst, the measured parameters.

Parameter pages generally also include reference parameters which are typically those of which most other parameters are a function. These are repeated on the various pages of each plant item, for example, the gearbox parameter pages all display the current value of shaft speed, variation of which affects most other significant gearbox parameters. In this case shaft speed is, in fact, an inferred measure of the true dominant variable, power.

Provision is also made on parameter pages for the display, in digital form, of the current values of selected parameters from a like piece of machinery e.g. the starboard gearbox parameter pages on a system destined for a two-shaft ship will include selected corresponding parameters from the port gearbox, and vice-versa. This allows correlation of trends, and is intended to promote the early detection of performance degradation in the plant concerned.

History pages, as the name implies, provide a window into the past behaviour of a parameter. In their current form, as shown in Figure 3, histories display sixty samples of parameter value on a rolling basis, i.e. as each new sample is added, the earliest of the preceding sixty samples is deleted. Most histories in the prototype system have a timebase of thirty minutes, and consequently the sampling interval is thirty seconds. Each history sample value is obtained by digital filtering of the values measured at the basic one second intervals.

The history facility allows an 'after-the event' assessment of conditions leading up to an incident which may have required corrective action via the control and primary surveillance system. This capability is a significant enhancement over the level of plant surveillance attainable with earlier 'gauge-and-panel' approach to the task.

The Logging facility is extensive, incorporating both automatic and demand logging. Automatic Logs are initiated either by time signals to give periodic logs, e.g. one-hourly, four-hourly, or twenty-four-hourly, or by the occurrence of an alarm condition on one of a number of pre-selected parameters to give incident logs. The periodic logs comprise groups of parameters to provide plant state overviews appropriate to the frequency of the log, while incident logs consist of parameters grouped according to their significance for a particular plant, the whole group being logged when any one of the parameters triggers the incident log.

Demand logs are initiated by request of the operator/maintainer, and effectively provide a 'note-pad' on which anything unusual in the way of parameter values may be recorded for further examination.

The logging facility calls into play peripherals additional to the display unit. The normal log output for onboard use is hard-copy via the printer, providing the basis for watch hand-over reports as well as compiling a performance record. The typical log output format is shown in Figure 4. The capability also exists to record log data on an appropriate magnetic medium to support automated onshore analysis of information.

The maintainer-assist facility provides an improved picture of plant performance compared with traditional methods of surveillance, by accessing parameters from a variety of sources, carrying out calculations with these parameters and with co-efficients stored within the system, and generating derived parameters for display. Typical derived parameters may include power, specific fuel consumption and efficiency.

Some basic analysis capability is incorporated in the maintainer-assist facility, at the present time in the form of life factor calculations on engine usage, to provide a better indication of the need to change a unit. As further methods of assessing plant life become available, it is hoped to incorporate appropriate techniques into the system. It is not the intention to attempt the eventual development of a system which will carry out all analysis and then output maintenance instructions, but rather to find new ways of presenting 'pre-digested' information so that analysis by the user is simplified.

Details of the system contents are given in the form of index pages, and access to the various facilities is obtained via hierarchical entry mechanism consisting of a grouped structure of plant types. In most cases the route into the system leads to a parameter page displaying the group or individual parameters of interest, and histories or demand logs are called up via the parameter page.

Additional facilities to be investigated in the next stage of the development include :

(a) A dynamic data recording facility to supersede the system on current ships. The present system utilises a UV recorder to obtain traces of dynamic parameter variation (e. g. during manoeuvres) and requires a tape recorder to extend its capability.

(b) The technical feasibility of incorporating into the overall system an onboard training simulator for the propulsion operator.

(c) The feasibility of producing a hard copy device attached to the VDU, either as a replacement for the printer or a complement to it.

IMPLEMENTATION

The system which provides the features described above consists primarily of a number of software tasks distributed amongst various processors. These processors, together with a range of peripherals, constitute the hardware of the system.

Software

The software tasks may be conveniently classified as either core software, which supports operation of the system, or applications software, which generates the user facilities. The core software consists of tasks such as Commap and Data Management, and these tasks, along with the operating environment which they provide under a Mascot executive, are discussed in Reference 4.

The applications software comprises the following main tasks :

(a) Data Assembly

(b) Logging

(c) History Recording

(d) Operator Interfacing

(e) Display Compilation and Servicing.

Each of these tasks will now be considered in turn.

<u>Data Assembly,</u> as the name implies, is concerned with the collection of plant information from various sources, and the organisation of that information to suit the requirements of the different user facilities. Although the data sources export scaled and normalised values in the form of engineering units, there are differences in the number formats as a result of the different processors (see discussion under Hardware) and translation of these formats is accomplished in the Data Assembly task. Also, a function of this task is the calculation of derived parameter values. These may then be utilised by facilities throughout the system exactly as any other parameter, with no special requirements for further processing.

<u>The Logging task</u> is triggered by various stimuli, e.g. signals from the ship's clock, and, in response, requests the values of the parameters appropriate to the particular log, calls up the format details and routes data to the required output device. Any conflicts arising from simultaneous demands for use of the logging facility are dealt with according to a pre-defined priority table, with temporary data storage being used to hold information for lower priority logs awaiting use of, say, the printer.

History Recording operates as an autonomous function which, on system initialisation, requests continuous updates of the values of all parameters designated as requiring histories. This task also implements the digital filter, discussed under Features, to ensure that signals are effectively band-limited and hence that aliasing errors are minimised. The sampled values are held on mass storage ready for recall and display when requested by the system user.

The Operator Interfacing task handles requests from the keyboard for particular information, and co-ordinates the operation of the system to provide user access, via the hierarchical entry mechanism, to the requested facility. This task also echoes keyboard entries on the display, and allows the user to correct any operator input errors before execution of the request.

The Display Compilation and Servicing task deals with the production of system output to the display unit, compiling information about static elements of the display from a number of sources in mass storage, and servicing the display with dynamic information about current values of the parameters. These data are passed via the graphics interpreter, which is an implementation of Webb Graphics, to the display unit. This task comprises a number of overlayable sub-systems, which are recovered from mass storage when a context change is necessary.

The system software is written in standard Coral 66, and documented according to the structure defined in Joint Services Publication (JSP) 188. This approach was adopted to support the aim of producing long-lifetime, maintainable software. A similarly structured format has been used for the documentation of other aspects of the system design such as user requirements definition and system hardware configuration.

Hardware

The main elements of the system hardware shown in Figure 5, are :

(a) Plant Control Units (PCUs) and Data Acquisition Units (DAUs)

(b) Central Surveillance Unit (CSU)

(c) Display Processor (DP)

(d) Visual Display Unit (VDU)

(e) Data Communications Links (DCL)

(f) Mass Storage

(g) Printer

(h) Magnetic Tape Unit.

The following paragraphs will discuss the equipment selected for implementation of these elements at this stage of the development programme.

The PCUs are part of the digital propulsion control system (Demonstrator) and are currently implemented with Ferranti F100L 16 bit microprocessors. A prototype DAU is being developed as a parallel activity within the main PASS development programme, and to evaluate the portability of the software this DAU is based on a DEC PDP 11/23 processor. The PCUs and DAUs constitute the data sources for the PASS system, and in a ship application will be mounted in machinery spaces, local to the plant.

The CSU comprises two processors, the Data Capture Processor and Main Processor, based on a PDP11/23 and a PDP11/34 respectively. The Display Processor(DP) is also based on a PDP11/23. Data Communications Links (DCL) between the data sources and the Data Capture Processor, which 'front-ends' the CSU, are handled by serial digital links of the Ferranti Serial Signalling System (S-Cubed) type. Othe inter-processor communications are also handled by S-Cubed links which can provide an information transfer rate of 3M bit/sec.

The capabilities of the 11/23 are required only for the Display Processor, and the decision to use 11/23 machines for the Data Capture Processor and Data Acquisition Unit was made primarily on the grounds of equipment commonality, to reduce spares holding and simplify system support.

The mass storage device chosen for the system is a 4.8M byte Winchester disc, selected with a view to providing a reasonably robust unit at low cost for evaluation system purposes. The type of mass storage to be used for shipboard application will depend upon the equipment policy to be adopted and the availability of interfaces to suit the preferred processor type(s).

The remaining items of hardware are all system output devices; the printer for hard-copy log output, the magnetic tape unit (cartridge type) for shore analysis, and the VDU as the main user peripheral. The VDU used on the system is an Interstate Electronics PD3000 gas plasma dot matrix panel, with alphanumeric and graphic capability. This device was originally selected for use on the submarine surveillance system programme and during the development of that system proved to be a high-quality display facility.

Distribution

The presently envisaged distribution of the software tasks within the processing power of the system is as follows :

(a) Data Capture Processor - the Data Assembly task runs in this processor, under a bare MASCOT kernel.

(b) Main Processor - the Logging and History Recording tasks run in this processor, under a MASCOT kernel hosted on a DEC RSX11M operating system.

(c) Display Processor - the Display and Operator Interfacing tasks run in this processor, under a MASCOT kernel hosted on a DEC RSX11S operating system.

This distribution is based on the capabilities of the selected processors and the present estimates of task sizes. Obviously changes in either of these factors could require re-distribution, the likely change in processor power when devices are selected for the shipboard application being an example. The stated aim of transportability of software is one of the prime reasons for using a task-based software structure.

Evaluation

The evaluation exercise which will follow the completion of this stage of the programme, will be aimed at assessing the performance of the system in functional terms. Page formats, accessing methods and range of information being displayed will be examined to determine their effectiveness in the shipboard application and the findings of this examination will steer the future development of the system.

CONCLUSIONS

This paper has described the way in which MOD and Y-ARD LTD. have been developing PASS, a comprehensive surveillance system for future generation Royal Navy warships. The aims of the system, in relation to the current standards of surveillance, have been shown to be :

(a) to reduce onboard manpower and/or skill level

(b) to improve operator effectiveness by assisting systems' understanding

(c) to minimise through life costs by promoting commonality and improving the flexibility of the initial design

(d) to provide a Maintainer Assist facility which will gather together information, calculating parameters as necessary from available data, and thereby ease the maintainers workload

(e) to minimise operator training requirements by making the PASS system as simple as possible to use.

Presently the project is nearing the end of Stage 1 of the 3 described. This has involved developing sufficient software,running on commercially available hardware, to prove the concept. Future work will be initiated once PASS is approved for a specific ship. Both MOD and Y-ARD LTD. have confidence that this development will provide a surveillance system highly suitable for fitting in future Royal Navy surface ships.

ACKNOWLEDGEMENTS

The encouragement given by Director General, Ships, and by the directors of Y-ARD LTD. is gratefully acknowledged, as is the generous assistance of the authors' colleagues. Opinions expressed are those of the authors.

REFERENCES

1. "A Structured Approach to Man-Machine Interface Design for Command and Control of Ship's Machinery". Lt. Cdr. J. L. P. Stainhausen, Ministry of Defence, J. N. Orton and J. P. A. Smalley, Easams Ltd.

2. "Objectives and Implementation of Future Warship Control and Surveillance Systems", Cdr. E. D. M. Floyd, Ministry of Defence, and J. B. McHale, Y-ARD LTD., (Paper to be presented at 6th Ship Control System Symposium, Ottowa, 1981).

3. "The Official Definition of MASCOT (Modular Approach to Software Construction, Operation and Test)", HMSO Publication, London.

4. "Machinery Control and Surveillance - Software Designed for Reliability and Maintainability". I. W. Pirie, Ministry of Defence and R. Foulkes, Y-ARD LTD. (Paper to be presented at 6th Ship Control Systems Symposium, Ottowa, 1981).

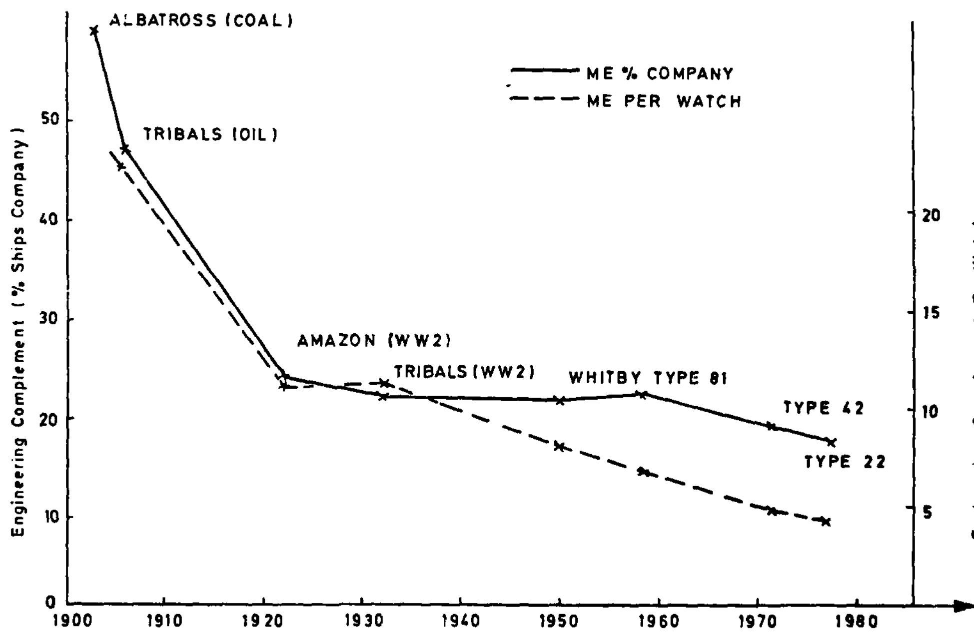

Figure 1 Engineering Complements On Successive Classes Of Destroyers And Frigates

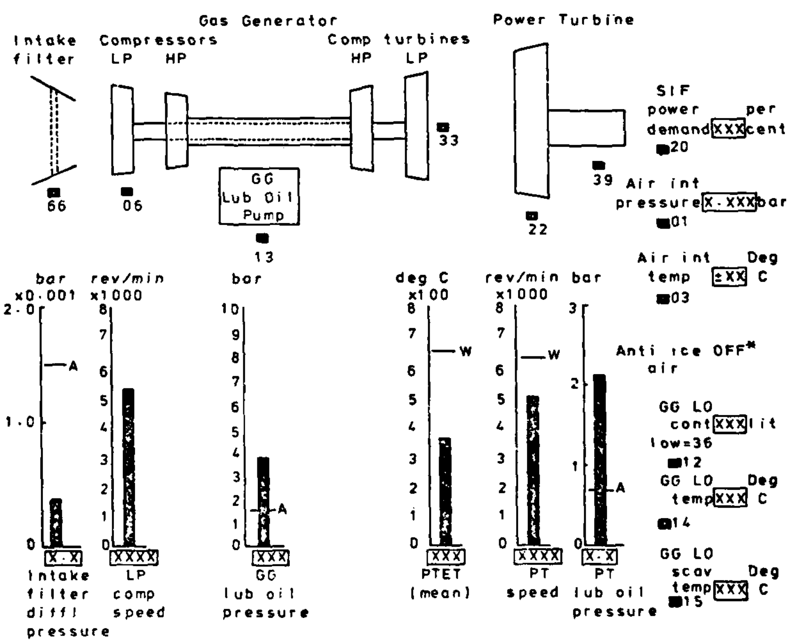

Figure 2 Spey Gas Turbine - Port Inner Engine

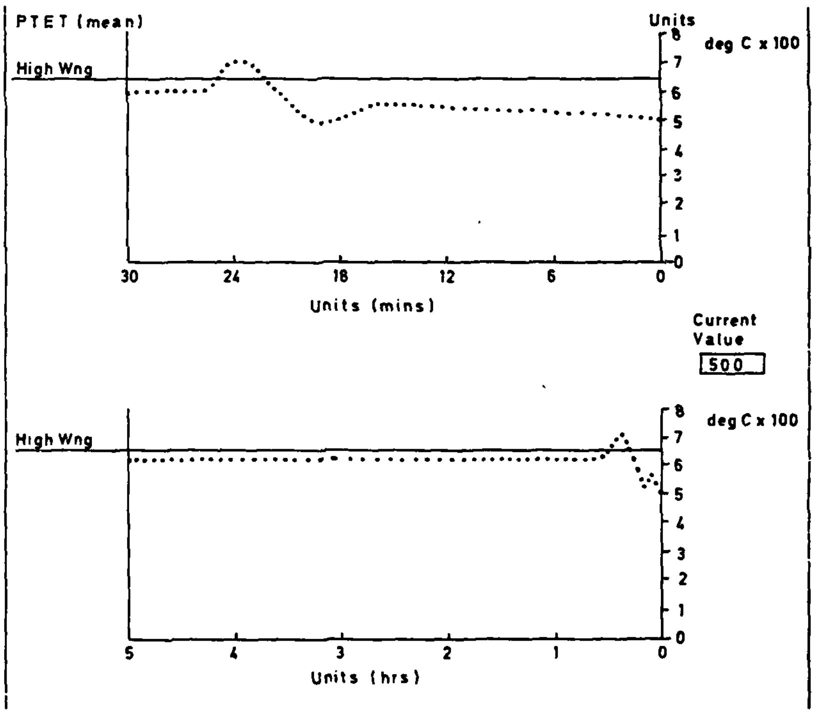

Figure 3 Time History Page

. . . . . .DEMAND LOG. . . . . .DEMAND LOG. . . . . .DEMAND LOG. . . . . .DE

SHIP: HMS DEMONSTRATOR

SHIP DATE 17 03 81 SHIP TIME 1321

GM DATE 17 03 81 GM TIME 1321

KEYBOARD REF NO			PARAMETER TITLE	UNITS	CURRENT VALUE	STATUS	LIMITING VALUE	CONFIDENCE LOW
06	01	01	Drain Tank Contents	per cent	94.0	-	90.0	-
06	01	03	Drain Tank Temperature	Deg C	78.0	High	75.0	-
06	01	05	GD Pump Discharge Pressure	bar	2.8	-	1.8	-
06	01	06	MD Pump Discharge Pressure	bar	0.0	LOW	1.8	-
06	01	07	LO Cooler Inlet Temperature	Deg C	78.0	HIGH	75.0	-
06	01	08	LO Cooler Outlet Temperature	Deg C	51.3	-	60.0	-
06	01	09	LO Filter Differential Pressure	bar	2.1	HIGH	0.5	x
06	01	10	Main Distribution Point Pressure	bar	2.7	-	0.7	-
06	01	12	GD Pump State	-	-	ON	-	-
06	01	13	MD Pump State	-	-	OFF	-	-

. . . . . .END OF LOG. . . . . .END OF LOG. . . . . .END OF LOG. . . . . .EN

Figure 4 - Typical Log Format

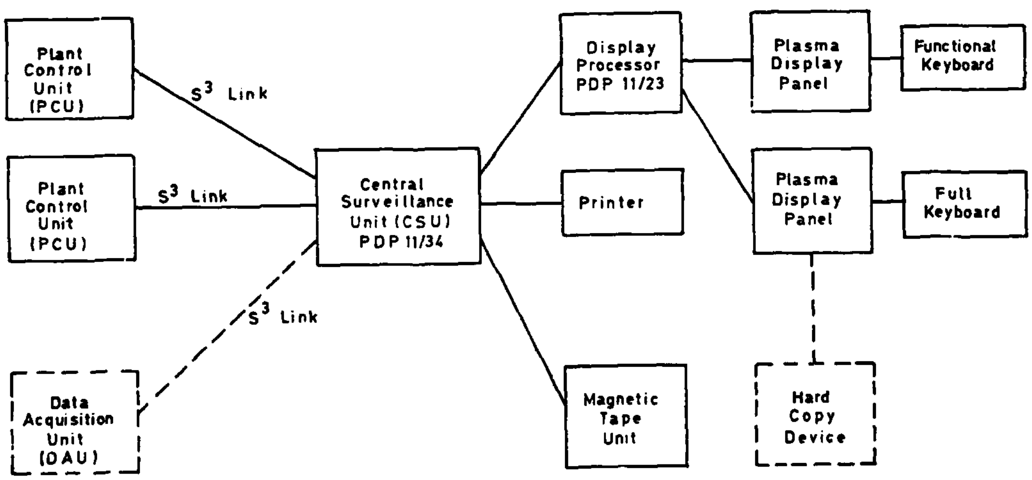

Figure 5 Propulsion And Auxiliaries Secondary Surveillance System (Pass)

AUTOMATION VERSUS MANPOWER REQUIREMENTS

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ABSTRACT

This paper examines the recent experience in the United States Navy where automation has been introduced into new ship designs. While other attributes are recognized in the introduction of automated shipboard systems, such as the ability to respond more quickly in combat situations, the paper focuses on the effects of automation upon ship manpower requirements. Specific examples are provided where expected reductions in manning were not achieved in recent ship designs where automation was incorporated for that purpose. While the use of shipboard automation is not without its critics, the U.S. Fleet appears to have accepted the concept. User feedback addresses the issues of reliability, the provisions for backup systems, the need for better qualified personnel and the concern about maintenance workload. The authors provide specific recommendations for improved guidance to ship designers as a means of more effectively applying automation in the ship design process.

INTRODUCTION

Continuous technological advances since World War II in equipment and ship design have created a Navy different in character and requirements from the "Old Navy," wherein operational goals were accomplished with sheer manpower. New systems and equipment now theoretically require fewer operational personnel than did their previous counterparts, but at the same time the need for more highly skilled maintenance personnel is increasing. Thus, in many instances, a situation is created where the combined operational and maintenance personnel demands of the new equipment exceed those needed for the less complex equipment.

Unfortunately, the available national manpower pool of people in a military age group is shrinking. This situation is further aggravated by inflation and by the competing manpower demands of the civilian sector of our economy. Thus, the increased manpower and skill requirements create an unacceptable situation for the Navy. During the 1960s and early 1970s, the Navy and the naval ship designers attempted to provide a solution to this problem through increased application of shipboard automation.

For the purposes of this paper, automation will be defined as the replacement of human supervision of machines and mechanized processes by automatic supervision.

While it is recognized that automation brings other benefits to the ship design, this paper limits its discussion to the effects of automation on manpower requirements.

In this paper, the authors will review the success that the ship designers have had in utilizing automation to solve the manpower shortage problem, analyze the present status of the automation concept, examine Fleet Acceptance of automation and draw some conclusions and recommendations for the future. A special effort will be made to address the need for specific guidance on automation that should be developed for the ship designers. This guidance should facilitate the determination of the proper level of automation that would be acceptable to the user and could be supported by available resources.

THE PRESENT STATUS OF AUTOMATION AND THE MANPOWER SHORTAGE

As mentioned before, during the last few decades new ship designs have been pursued which seek to minimize the overall manning requirements through extensive use of automation. The LHA-1, DD963, CGN-38, and the FFG-7 Class ships provide a good example of this trend.

The designers did realize that these ships would require more highly skilled individuals to operate and maintain them. At the same time, they thought that the total number of personnel aboard would be reduced, alleviating the manpower shortage problem. This goal, however, was never achieved. The total manning numbers have remained approximately the same. The only change has been the increase in the skill levels required.

During this era, the designers also attempted to pursue new design concepts that would reduce the required ship manning to a level of a space capsule equivalent. The Essential Manning Concept (ESMA) pursued in the early 1970s by NAVSHIPS, proclaimed that it was feasible to operate a destroyer with a crew of between 15 to 62 men depending on the level of automation involved.[1] Another study, Project Chameleon,[2] conducted by students of the U.S. Naval Academy, concluded that a 2000-long ton displacement destroyer could be designed for the post-1990 time frame with a crew of 25 highly skilled individuals. Obviously, tremendous reliance had to be placed on very advanced concepts of automation.

Some voices of caution warned that perhaps automation was not the ultimate solution to the manpower shortage since automation might require more skilled personnel. For example, Capt. C.A.L. Swanson, USN, in an address to the American Society of Naval Engineers in 1972 cautioned that "...over-sophistication generally leads to complexity with attendant poor reliability and excessive support requirements."[3] In the long run, skilled individuals are more costly than the unskilled personnel that they replace. In addition, requirements aboard a man-of-war for battle conditions, such as damage control, had to be satisfied under any circumstances. For example, even though operational manning could be reduced by 15 to 25 men for a destroyer, to satisfy the minimum damage control requirements, another 25 men (or more) may be needed on the ship. Thus, the overall effect of the savings due to automation would be nullified. In this case better savings could be achieved with less automation and retention of selected manual functions. Of course, this combination would have to be carefully evaluated during the ship design phases.

LT Robert A. Ortman, USN, while completing his thesis for a postgraduate degree in 1976, examined the question of reduced manning vis-a-vis our present capability to automate ships.[4] He concluded that a 120 to 160-man range provided the best combination of personnel and equipment that would meet the requirements of an improved FF 1052 type ocean escort (gas turbine propulsion with other highly automated features) with a low amount of acquisition risk. This range is about 108 to 148 men less than the 268 men called for in the 1976 Ship Manpower Document (SMD) for the original FF 1052 Class. To achieve this, automated propulsion control was required, combat systems had to be integrated, reduced bridge manning introduced, and gun and missile systems remained unmanned during wartime cruising conditions. If fewer men were to be assigned, the ships would not be able to carry out all assigned missions. LT Ortman believes that manning with 15 to 25 men is unlikely to become reality within the next decade or two. Rather, a gradual approach to "crew reduction" efforts should be taken.

Looking at actual results achieved, we can observe that the general trend has been towards an increase in crew size rather than a decrease. Table 1 illustrates this trend by depicting the initial design manpower estimates for some new ship classes and also showing the latest manning requirements for the same ships. Each class demonstrates an increase in the manpower requirements. This growth varies between 4.7 to 30.5% (DD963). Thus, one may ask, what did happen to our anticipated reductions? They seem to have disappeared.

Table 1. New Ship Manpower Requirements.

	Final Design Estimates (Date)			Actual Manning			%
SHIP CLASS	OFF	ENLISTED	TOTAL	OFF	ENLISTED	TOTAL	INCREASE
CGN-38	27	445	472(3/76)	28	507	535(7/79)	13.3
CG-47	20	275	295(12/75)	23	319	342(3/81)	16.0
LHA-1	50	681	731(7/74)	51	837	888(3/78)	21.5
DD963*	18	224	242(5/76)	19	297	316(6/81)	30.5
DD993*	30**	287**	317**(1/79)	20	312	332(3/80)	4.7
FFG-7*	10	148	158(12/72)	12	180	192(8/80)	21.5

* w/o LAMPS Detachment
** Imperial Iranian Navy (IIN)

The following facts seem to have contributed greatly to the overall manpower increases:

- o Additional systems/equipment added during the ship design/building phase;
- o Over-optimistic low assessment of the manpower requirements for new systems/equipment (this includes manufacturer predicted maintenance);
- o Inability to accurately estimate the total maintenance workload since many items are introduced by the builder as contractor furnished equipment;
- o Inability to identify mandatory manual backup requirements; and
- o Unavailability of modular replacement parts necessitating onboard repairs.

About 50% of the growth illustrated in Table 1 can be attributed to increases from manning underestimates generated with anticipated savings due to automation. Additional inaccuracies are due to low manning estimates of the overall maintenance workload and the insufficient initial manual backup features. In many cases, automated systems were expected to reduce the required number of operators and also decrease the maintenance workload. Unfortunately, while automation did reduce the number of operators, it also increased the number of skilled technicians required

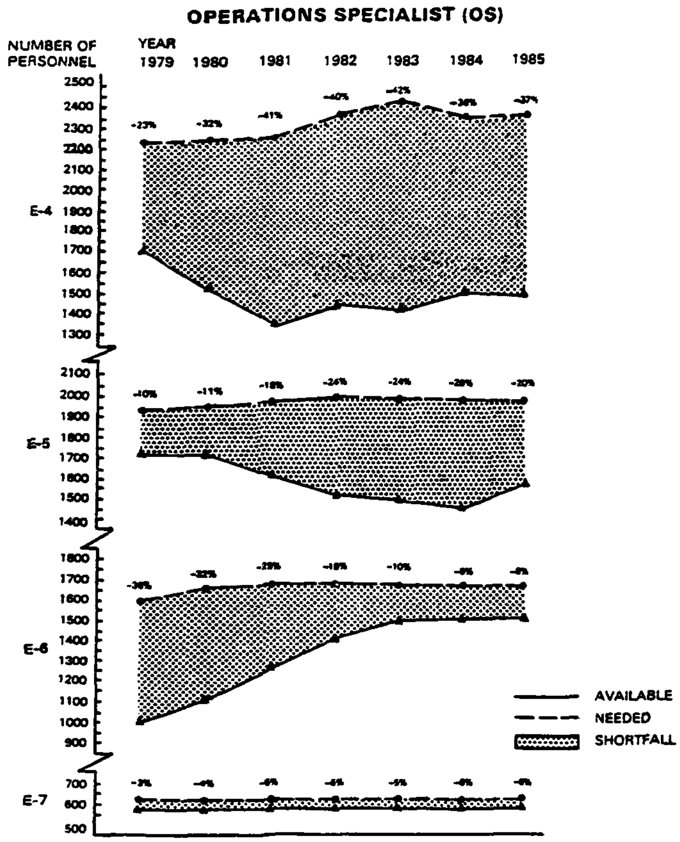


Figure 1. Operations Specialist (OS).

ENGINEMAN (EN)

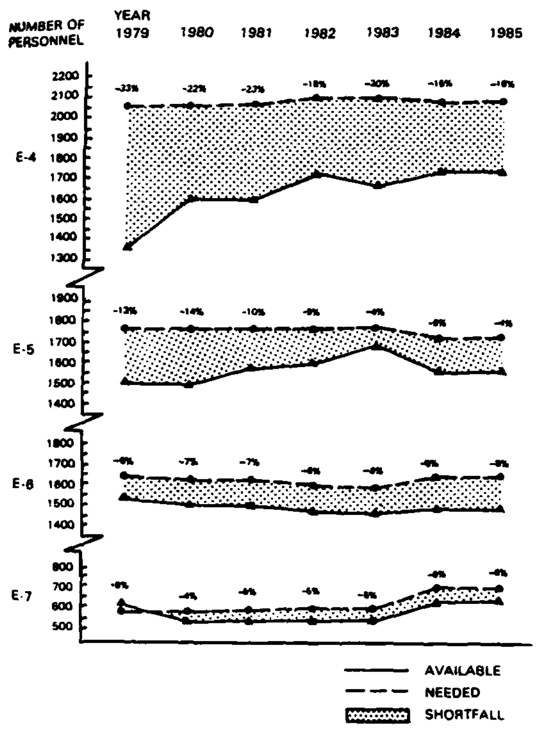


Figure 2. Engineman (EN).

FLEET ACCEPTANCE OF AUTOMATION

An important factor that a ship designer must consider is the concept of Fleet Acceptance of automation. As mentioned earlier, the Navy has had an extreme interest in shipboard automation. The interest has not been merely a fascination with new electronic gadgetry but rather one that has been pursued in the interest of doing the job better. The authors, having had the opportunity to observe Fleet reaction to the introduction of new equipment and systems, have come to the conclusion that automation in itself is widely accepted and, in fact, strongly supported.

Advocates of Fleet automation frequently emphasize the necessity to analyze massive amounts of data input and to respond correctly in a matter of seconds. They are convinced that these results can only be achieved through automation. More recently, with manning being a concern, this interest has been extended towards seeking ways to make shipboard evolutions less labor intensive. But is there another side to the question of whether automation is accepted by the Fleet? Obviously there is. The Fleet wants the benefits of responsiveness and less manning but they do not want unreliable equipment or systems that cannot be depended upon to function in wartime conditions or that require more maintenance than before.

Several years ago, while a new small carrier (CVV) was being designed, Fleet representatives were invited to attend design reviews very early in the ship design process. Notable among the Fleet comments received was a common concern of ensuring that new equipment was thoroughly tested and proved prior to incorporation into the design. The Fleet made it clear that they desired the capabilities but they also wanted them with guaranteed reliability and immediately available for use. There were preferences for semi-automated systems as opposed to fully automatic in several cases. The sailor assigned to a job aboard a ship appreciated anything that was done to make his job easier and did the job better, but he didn't always sense that this was so in some of the so-called automated features being placed aboard his ship. The sailor's outlook differs from that of a Navy strategist, whose primary objective is to maintain a competitive edge over potential enemies with more sophisticated equipment as the solution.

In February 1980, a study of lessons learned in SPRUANCE Class (DD963) destroyers was published and included feedback from the Fleet. Significant discussion with respect to automation and ship manning is contained in the following excerpt from the report:

> "An important consideration in the development of the DD963 SMD (Ship Manpower Document) was that initial planning diverged from traditional concepts (e.g., reduction in watch stations, unmanned engineering spaces, etc.). Although the concepts and plans were reviewed by the Fleet and Type Commanders, when the ships reached the Fleet practicality eventually dictated that manning levels be increased to account for actual workload."(9)

This statement implies that either the applications of automation did not achieve the desired reduction in manning and/or a simple lack of confidence or acceptance of concepts existed.

In brief, there is no Fleet Acceptance problem to the introduction of automation into ship control or other systems if it is understood by the user. The designer's intent in the development of the automated features has to be conveyed clearly to the user via some kind of documentation. There is a universal agreement that these systems, like any other, must be adequately supported and maintained. Both functions involve properly trained and qualified personnel. Whether this can be reflected in the design process and subsequently achieved is an issue.

DESIGNING FOR AUTOMATION

The most logical place to begin ensuring that new designs are supportable is during the design of the system. This will require additional design guidance which addresses supportability. The guidance development could begin by establishing some baseline do's and don'ts based on the Fleet's experience with automated systems.

The design process must also be expanded to include Fleet involvement early in the concept design. The process must include all system level parameters affecting the system's operation. This could be accomplished at the beginning of the design process through automation impact assessments conducted in areas considered sensitive (i.e., key parameters) such as operability, cost, and other factors at the man/machine level. These assessments can be measured against predetermined values (Figures of Merit) which have been determined acceptable and are available as guidance. Figures of Merit for the specific design can then be determined and used to drive the design toward acceptability.

Utilizing Fleet's Experience

It is a well-known fact that system evaluations are different for lead and follow-on ships. Evaluations are divided into a prototype period, the year of operational testing which any new major design requires to either iron out all the wrinkles or debug the system, and the subsequent period, which the writer calls the "this is as good as we can make it" time frame, during which improvements cannot be made without entailing major changes. This situation can be exemplified by referencing the LHA-1, which the writers consider the prototype, and LHA-3, which depicts the "as good as we can make it" period. Given the same systems and following the post shakedown corrections on the lead ship, the degree of Fleet Acceptance was much higher for the follow-on ships versus the lead ship. The latter is the realistic period to measure the Fleet's experience.

Preliminary feedback indicates that had Fleet guidance been utilized during the design process and had it followed some instructional procedures, it could have contributed greatly to the elimination of significant disparities between initial ship manning requirements versus final validation manpower requirements.

Thus, in order to determine and introduce an acceptable level of automation in a new ship design, design guidance (i.e., manning criteria) and implementing instructions must be developed by the Command and implemented by the design codes into the design process. This guidance should be developed and introduced into the design process in such a fashion that the ship personnel, equipment, procedures, and software are treated as an integral part of the total ship design as opposed to the ship as the sole consideration.

System Evaluation

To clarify the intentions of the authors in discussing future recommendations, a typical case will be discussed.

The functional goals of a typical system will be presented together with a reference base which shows where we are coming from or upon what we are trying to improve.

One typical system which has a great deal of feedback data available is the gas-turbine setup designed for the DD963 Class. This particular system can be divided into two major subsystem categories:

1. Ship Control and Navigation, which was designed to increase operational efficiency, improve command decision capability, direct conditions, and improve flexibility; and

2. Engineering Control, which is further subdivided into two areas--main propulsion machinery and auxiliary and electrical machinery. The system was designed to provide direct monitoring and control capabilities from a Central Control Station (CCS) for the propulsion, auxiliary, electrical, and damage control systems. As backup, each engine room contains a Local Operating Station for display and local control of its corresponding propulsion plant. Remote sensing is used to monitor machinery in the auxiliary machinery rooms, which includes electrical machinery.

The gas turbine propulsion system of the DD963 Class can be (and was) compared to the steam propulsion system of a similar ship, the DDG-2 Class, then in Fleet use.

The new systems provided:

- o Remote starting for each gas turbine from a CCS which is automatic and sequential;
- o Automatic main clutch engagement and remote control of disengagement from CCS;
- o Fully automated central controls (including casualty control) operation of the propulsion machinery;
- o Remote integrated control of propellers RPM and pitch for each plant for the bridge or the CCS; and
- o Automatic bell and data logging.

These features were designed to minimize manning requirements by reducing watchstanders and to facilitate safe, reliable, and efficient operation of the propulsion machinery.

A postdelivery investigation of the DD963 machinery plant automation effort has been documented by Fitzpatrick.[10] The findings delineate the following design intentions and document the results achieved:

- o "The automated features, including pilot house/bridge control, are regularly used and generally have operated reliably.
- o The DD963 Class ships are operating with reduced rated and non-rated ratios in propulsion plant rates from those required at the Fleet level.
- o Given current manpower, accomplishment of PM, CM and FM is below requirements. This implies an unrealistic workload reduction effort.
- o The ships are operating with more watchstanders than predicted during design. Unmanned operating stations were being manned to adequately cover ship safety.
- o Operators had no training or the means (system interface documentation) to understand totally the consequences of some operator actions.
- o Local manual control of the CPP system through mechanical linkage is not provided.
- o No backup electronic control unit for the gas turbines is provided.
- o The maintenance requirements for the propulsion plant controls were poorly estimated."

GUIDANCE DEVELOPMENT FOR SHIP DESIGNERS

An assessment of ship propulsion automation systems was completed by the Ship Systems Engineering Management and Material Office, Ship Systems Directorate, Naval Sea Systems Command in May 1979. This study determined the impact of existing automated, centrally controlled systems with respect to Fleet Acceptability, and provides a detailed criteria assessment of individual automatic and remote control features. It did not identify the source of the problem nor did it develop criteria and guidance which the automation systems design process needs. It should, however, be used as a source of assessment data.

No quick and easy solutions are presently discernible and, if they were, a high degree of risk would still exist until the cause of the problem could be determined. Having identified the cause, guidance to reduce future problems should be developed. This is consistent with the major goal expressed by the Chief of Naval Material in his letter of 25 October 1977: "Improve the match between new equipment and systems introduced into the Fleet and the Fleet's ability to properly operate and maintain such material...."[8]

Identifying the Cause

To analyze the cause of a problem, a perusal of Operational Evaluations, Technical Evaluations, INSURV Report, Special Inspections, Casualty Reports, and Navy Fleet Material Support data should be made to identify problem areas requiring analyses.

Concurrent with this, a compendium of currently available design guidance, specifications, standards, and other design criteria utilized in the design process should be compiled. This will serve as the design guidance baseline document and could be analyzed to determine adequacy.

The analysis of this design criteria and guidance should include, as a minimum, design and performance requirements, ship characteristics and operational requirements, top level requirements, top level specifications, ship specifications, and system and subsystem specifications. The assessment should also cover design studies, ILS plans, maintenance/repair philosophies, manpower constraints and policies, personnel and training plans, preliminary ship manpower documents, and ship manpower documents.

Following these analyses, plans should be made to visit Atlantic and Pacific Fleet Staffs for first-hand assessment of selected shipboard automation identified as problem areas. The assessment would be used to determine key parameters and establish figures of merit (FOMs) for those parameters to measure the Fleet Acceptability of future automated systems. The following parameters should, as a minimum, contain FOMs:

- Maintainability
- Supportability
- Operability
- Necessity
- Personnel availability
- Training impact
- Manpower impact
- Overall effectiveness

These baseline FOMs could serve as sensitivity guidance for similar new systems. Key parameters identified during the assessment should be analyzed before the design begins to establish, on a case basis, specific FOMs tailored for each new system for determining an acceptable level of new automation. Non-critical factors should be assigned a measure relative to their importance. It has already been presupposed that the problem was directly related to inadequate criteria; however, by using the results of both preceding efforts, the following questions can be answered:

- o How many problems were caused by the procurement strategy (i.e., Concept Formulation/Control definition during previous SECNAVs)?
- o Did those responsible for the design of the systems follow the design criteria and guidance applicable to the active Fleet automated systems?
- o Were current problems caused by inadequate criteria?
- o Did the automated systems receive an adequate design review?
- o Were differences between the "as-built" automation system and the "as-designed" version responsible for problems subsequently encountered?
- o Was the design (intra & inter) documentation adequate?
- o If the designs were adequate and Fleet acceptable, was the Fleet improperly trained or indoctrinated?

Applying the Methodology for Determining an Acceptable Level of Automation

Emphasizing the key parameters, a design procedure should be developed to call out areas of design requiring impact study at the man/machine level at the beginning of the design process.

Guidance for the preparation/execution of impact studies of key parameters should be developed by the functional codes to be used with detailed supporting information developed by the Command. The instructional document should indicate when an impact study is necessary to establish FOMs for acceptance.

These FOMs should be computed based on impact studies for each proposed system and be kept in the design history notebook for use as budgets during the design and later during the system evaluation. These FOMs should be reviewed by a Fleet representative for concurrence (see Table 2). The design procedure should contain the baseline FOMs for types of systems and also the format for the impact study. It will serve as guidance for high level automation decisions.

Automated systems which do not meet minimum FOM requirements for sensitive areas should be redesigned. If a decision is made to waive the evaluation, however, there must be written approval and the assessment or impact of the failed area well documented. This assessment should not become buried in the design history. Lessons learned must be documented in the design history and result in updated design criteria and guidance documents. These documents will drive the design process and provide the means to reduce Fleet unacceptable systems.

Parallel to this guidance development, the Command should establish a capability for providing accurate manpower supply projections early in the design/development process.

A formal directive on shipboard automation should be prepared to furnish cognizant system designers with guidance for automating shipboard systems. This instruction should also assign the responsibilities and establish the procedures and reporting requirements during ship design. (See Tables 3 and 4.) The instruction should delineate procedures for identifying qualifying areas for impact studies and a scheme or methodology to establish and measure the acceptable level of automation for systems.

Table 2. (1) Ship Control System.

(2)	(3) Key Parameters*	(4) Threshold & Critical Index	(5) Grade	(6) Remarks
	Cost	1 - 10	1	Acquisition Cost Only
X	Maintainability	0, 3 - 10	3	Must have quick access to selected machinery
X	Supportability	0, 3 - 10	3	
X	Operability	0, 5 - 10	5	Conditions, I, II, III, IV
	Necessity	1 - 10	1	Needed for Reaction Time Improvement
X	Personnel Avail.	0, 3 - 10	0	
X	Training	0, 3 - 10	3	
	Manpower	1 - 10	1	
X	Manual Backup Remote	0, 7 - 10	7	
X	Manual Backup Local	0, 6 - 10	6	
X	Safety	0, 5 - 10	5	
	Threshold Acceptance	38	35	System is below acceptable value

SYSTEM EVALUATION

Table Descriptors

(1) & (2) Enter mark if area (item 2) has been determined by the procedure to be qualified for an impact assessment before a grade is given. This qualifying procedure is called out in a NAVSEA instruction.

(3) Pertinent design considerations (key parameters).

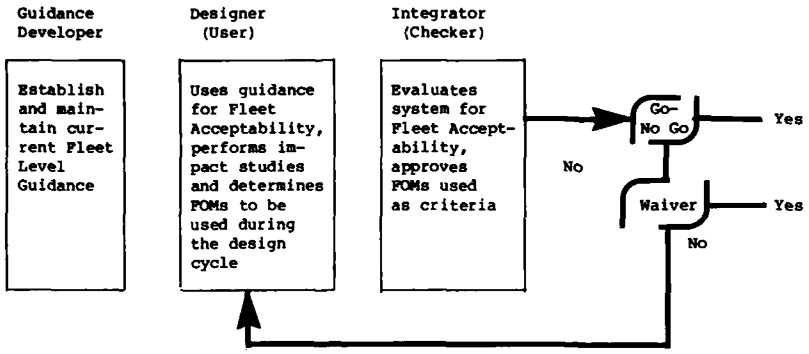
(4) Critical Index determined through an impact study which identifies a FOM of acceptability. An index of 0, 5 - 10 means that below 5, it has been determined through analysis as being unsatisfactory and the entire system will fail (below threshold of acceptance).

(5) Grade assigned to all areas. Areas requiring impact assessment, however, must have an assessment report before any grade can be given.

(6) Self-explanatory.

* All key parameters have been treated equally for simplicity.

Table 3. Cognizant Functional Organization.



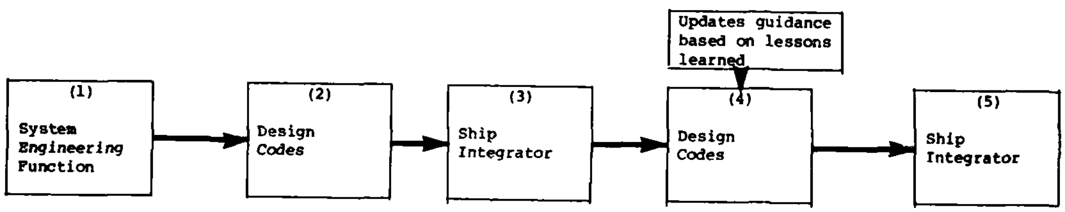
Duties

(1) Develops and maintains current Fleet Acceptability guidance, criteria, and implementing instruction. All Design Criteria established with the cognizant code as lead. Lead determines where and how in the design process this criteria will be used.

(2) Using the guidance and design criteria, selects qualifying areas for in-depth analysis to satisfy that the impact in these areas meets an acceptable level.

(3) Using the same guidance and design criteria, determines that all qualifying areas have been analyzed in-depth and enough information is available to perform an evaluation. (Assigns a critical index weight, checks FOMs and gives a grade).

Table 4. Start-up Sequence.



(1) System Engineering Function	(2) Design Codes	(3) Ship Integrator	(4) Design Codes	(5) Ship Integrator
Develop guidance criteria and implementing instruction for new designs. Identify key parameters. Assign baseline FOMs for system types.	Design codes implement guidance and criteria in the design cycle.	Integrator calls out the instruction for using the guidance and criteria during design of equipment designated for specific ships.	Design codes use guidance criteria and perform impact studies on qualifying areas according to the instruction. Determine FOMs for key parameters, to be used during the design.	Determines if system is ready for evaluation and performs an evaluation. Failed systems will either be reworked by the design codes or a waiver asked. In either case, the evaluation will be well documented as design history and reason for for failure are determined by the System Engineering Function.

SUMMARY

In summary, one can conclude that automation is not a simple solution to our present and future manpower problems. This has been proven over the last few decades where automation was introduced in our new ship designs. Automation may or may not reduce manning requirements. It definitely requires, however, higher skill demands for the new systems. Our experience with ships like the FFG-7, DD963, LHA-1, and others have proven this aspect. Fleet Acceptance is also an issue that must be considered. The authors feel that a guidance system, as described in the paper, must be provided to the ship designers to avoid some of the pitfalls that we have experienced on our recent ship designs. This will be a great effort but it will at least optimize the use of our scarce manpower and financial resources.

REFERENCES

(1) W. Lawrence Fulton, II, LCDR, USN, "Essential Manning - Its Impact on Destroyer Design, Operation and Maintenance," Naval Engineers Journal, Vol. 86, No. 3, June 1974.

(2) D. Crowder, ENS, USN, et al., "Project Chameleon - A Multidisciplinary Student Design Study" Annapolis, U.S. Naval Academy, May 1974.

(3) C.A.L. Swanson, CAPT, USN, "Design Considerations for the Future Navy," Naval Engineers Journal, Vol. 84, No. 3, June 1972.

(4) Robert Ortman, LT, USN, "The Impact of Reduced Manning on Naval Ships," Unpublished thesis for partial fulfillment of the requirements for the degree of Ocean Engineer and M.S. in Shipping and Shipbuilding Management, Massachusetts Institute of Technology, May 1976.

(5) E.A. Koehler and M.A. Miller, "Manpower Availability - Navy Enlisted Projections - FY 79-85," Report Number NPRDC SR 80-5, Navy Personnel Research and Development Center, San Diego, CA, December 1979.

(6) Department of the Navy "Military Manpower versus Hardware Procurement Study (HARDMAN) Final Report," Office of the Chief of Naval Operations Publication ser 96/90273, Washington, D.C., dated 26 October 1977.

(7) Dynamics Research Corporation (DRC) "Application of the HARDMAN Methodology to the Guided Missile Destroyer (DDGX)" 2 Vols., DRC Report R-371U, Wilmington, MA, June 1981.

(8) Chief of Naval Material letter to Commander, Naval Sea Systems Command, Subject: "Shipboard Automation," dated 25 October 1977.

(9) SPRUANCE DD963 Class Destroyers Lessons Learned, DD963 Project Office, Naval Sea Systems Command, February 1980.

(10) E.E. Fitzpatrick, "Propulsion Plant Automation Review," Naval Sea Systems Command, Washington, D.C., May 1979.

SHINMACS MACHINERY CONTROL CONSOLE DESIGN

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ABSTRACT

The Defence and Civil Institute of Environmental Medicine (DCIEM) has completed the design phase of a machinery control console development project in support of Canadian Forces maritime requirements.

This paper describes the development of machinery control consoles for SHINMACS (Shipboard Integrated Machinery Control System) with respect to:

- tasks performed during the design phase;

- application of a human engineering design process;

- analyses of operator information and control requirements; and,

- descriptions of SHINMACS console components, control functions, and displays and their operation.

INTRODUCTION

A previous paper, presented at the Fifth Ship Control Systems Symposium (Ref. 1), reviewed existing machinery control console designs in terms of a number of human engineering considerations including grouping of display instruments, usage of graphic displays, patterning of displayed information, access to machinery data, and implementation of integrated and predictive displays. It discussed the impact of propulsion control technology on machinery control console design and presented a number of human engineering advantages inherent in the utilization of computer graphics displays for presentation of propulsion system information to machinery operators. The prime advantage is, of course, that system information can be presented in task-related chunks and , within chunks, can be formatted in process flow and mimic diagrams to reduce operator cognitive workload. (The human operator processes information using what are called cognitive processes in order to detect signals, recognize patterns, selectively switch attention, solve problems, make decisions, store and retrieve data from memory and control movement.) Human engineering design of SHINMACS Machinery Control Consoles will reduce task and procedure learning times for trainee operators and increase the effectiveness and reliability of trained operators under conditions of combat stress such as, for example, extended watches during sea battle.

Project Background

The Defence and Civil Institute of Environmental Medicine (DCIEM) has provided human engineering support to the Canadian Forces for development of a number of maritime systems. In 1977 DCIEM was tasked to undertake a human engineering investigation of ship propulsion control console design for a microprocesor-based, distributed control system called SHINMACS. The prime objective of this design project was to develop human engineering design requirements for SHINMACS consoles. In order to achieve this objective, DCIEM undertook and completed the following tasks:

- study of existing machinery control concepts;
- examination of existing machinery control console (MCC) design concepts;
- familiarization with DDH-280 machinery system components and operations;
- familiarization with the proposed SHINMACS technical specification;
- review of the state-of-the-art of electronic display and computer input device technologies;
- review of fundamental concepts in man-computer interaction;
- analysis of watchkeeper/watchsupervisor functions and tasks in ship machinery control systems;
- application of relevant human engineering design criteria to the development of preferred SHINMACS machinery control consoles, including full-scale mockups.

Human Engineering Design

The development of SHINMACS machinery control consoles has taken place within the framework of a human engineering man-machine interface design process. Briefly, this process involves:

- analysis of operator functions and tasks;
- specification of operator information and control requirements;
- specification of console displays and controls, including their layout and interaction;
- specification of console dimensions and finish, and of access to modules;
- construction of full-scale mockups and/or simulations to evaluate console layout, functions and operations.

The first four steps in this process have been completed. Full-scale mockups of the main and supervisory consoles have also been completed (see photographs in Figs. 1-7).

Comparative analysis of computer display and conventional process control instrumentation technologies clearly demonstrated that the capabilities of electronic display systems could be used effectively in presenting information to SHINMACS operators. Therefore, a decision was made early in the conceptual design stage to implement electronic, specifically CRT, display systems in design SHINMACS machinery control consoles.

Baseline Configuration

Although SHINMACS, including its consoles, has been designed for application to a wide variety of ship machinery system configurations, it has been necessary to assume some basic controlled machinery configuration for purposes of human engineering development. Therefore, the DDH-280 propulsion, ancillary and auxiliary systems have been adapted as the baseline configuration.

The DDH-280 propulsion plant consists of two main and two cruise gas turbine engines driving two reversible pitch propellers through synchronous self-shifting clutches along two shafts in a COGOG (Combined Gas or Gas) arrangement. The two shafts are independently controlled.

This baseline configuration is evident in the layout of control panels and CRT page formats for the two SHINMACS consoles presented in this report.

SHINMACS Machinery Control System

Although a comprehensive description of SHINMACS is available elsewhere in these Proceedings, a brief summary is presented here for review purposes.

SHINMACS is a microprocessor-based distributed control system which provides automatic control and monitoring of ship machinery systems. (Most NATO navies are currently developing similar fully microelectronic digital control systems.) In SHINMACS, microprocessors are distributed througout the ship in close proximity to controlled plant and communicate via a high speed serial data bus. Plant status data, including high-level alarm messages, are sent by on-plant processors to other processors handling data logging and display tasks. Command inputs at consoles on the bridge or MCR (Machinery Control Room) are processed by these latter processors and control messages are sent in turn to the on-plant microprocessor controllers. It is from the MCR that all main and ancillary propulsion systems and auxiliary systems are controlled and monitored.

OPERATOR INFORMATION AND CONTROL REQUIREMENTS: A SUMMARY

Machinery Watchkeeper

The information and control requirements of the watchkeeper are distinct from those of the watchsupervisor. The primary information requirement of the SHINMACS watchkeeper, in his role of supervisory controller, is for display of a small amount of highly integrated, high level information about overall machinery system status. Priority is given to main propulsion systems (i.e., engines, clutches, gearing, shaft bearings and propellers).

Primary information is organized into overview displays of:

- propulsion plant and shaftline component status;
- ancillary and auxiliary system status;
- current alarms;
- watchkeeper-selected parameters.

Complementing the primary information requirement is a secondary requirement for detailed status information about each subsystem in the main propulsion systems and for each ancillary and auxiliary machinery system.

The primary control requirements of the watchkeeper are for:

- control of shaft knots in automatic propulsion control mode;
- control of engine power and propeller pitch in manual propulsion control mode;
- operation of all remotely controlled ancillary and auxiliary system components;
- control of propulsion control mode and station-in-control;
- selection of information for display.

Machinery Watchsupervisor

The primary information requirement of the watchsupervisor, in his role as supervisory , or executive, monitor, is for display of data from:

- machinery log database (i.e., scheduled logs);
- demand logs (i.e., long term logs);
- dynamic data analyses (i.e., short term logs);
- health monitoring systems (i.e., gas path analysis, bearing wear monitoring, and vibration monitoring).

These data are not normally available to the watchkeeper. The secondary and no less important information requirement of the watchsupervisor is for display of all information that is available to the watchkeeper.

Watchsupervisor primary control requirements relate directly to selection of parameters for the compilation and display of data from logs, dynamic analyses, and health monitoring systems. His secondary control requirements are for:

- control of propulsion machinery in a backup role;
- control of ancillary and auxiliary machinery in a backup role;
- control of propulsion control mode and station-in-control in a backup role.

It must be emphasized that the watchsupervisor's main requirement is for control and display of data required for supervisor monitoring. The requirement for backup control is considered to be of lower priority and controls and displays for that function are specified only for the case in which the watchkeeper is unable to exercise his control function by virtue of the inoperability of his workstation, the Main Machinery Control Console.

SHINMACS CONSOLE DESIGN FEATURES

The information and control requirements of the watchkeeper and watchsupervisor described above have dictated the specification of two consoles:

- the Main Machinery Control Console for the SHINMACS watchkeeper (Figs. 2-5);
- the Supervisory Machinery Control Console for the SHINMACS watchsupervisor (Figs. 6 and 7).

  Detailed design specifications for both consoles are given in References 2 and 3.

Main Machinery Control Console (MMCC)

The MMCC is a single-operator, three-CRT workstation having controls for:

- automatic and manual control of propulsion plant;
- selection of propulsion control mode and station-in-control;
- control point command option selection;
- display page selection;
- page assignment;
- cursor control and control point selection;

- alarm acknowledgement;

- engine start enable and direct hardware trip;

- voice communications;

- CRT monitor brightness and contrast.

The primary information requirement of the watchkeeper is provided by four overview display pages:

- the Propulsion Overview;

- the System Overview;

- the Alarms Overview;

- the Operator Monitor.

Propulsion Overview:

The Propulsion Overview (Figs. 8-10) is a graphics display of shaftline status and contains the following:

- mimic diagrams for both shafts showing engines, clutches, gearboxes, shafts and propellers;

- analogue and digital display of the four major parameters for each driving gas turbine engine;

- percent power for all four engines;

- propeller pitch and RPM for both shafts;

- order, reply and actual knots for each shaft;

- engine power control signals (cruise and main) and propeller pitch control signals when in manual propulsion control mode;

- propulsion control mode and station-in-control;

- control transfer line menu (upon demand);

- the number of active alarms;

- status of the 'battle override' option;

- sea state.

The purpose of the Propulsion Overview page is to give the watchkeeper and the watchsupervisor a continuous and comprehensive integrated display of main propulsion system status.

System Overview:

The System Overview (Fig. 11) is simply an annunciator type display of ancillary, auxiliary, and electrical systems status.

Each system is represented by a block which displays the system name in normal video if the system status is normal or in inverse video if the system is in alarm status. The operator can quickly summarize from this type of display format which systems are in alarm.

Alarms Overview:

The Alarms Overview (Fig. 12) is a list of alphanumeric alarm and warning messages with the most recent alarm displayed near the bottom of the screen. The list is scrolled upwards with the addition of new messages. When the list reaches the top of the screen it overflows onto a second page. Messages on the second page can be recalled by scrolling the list downwards. The watchkeeper may order the list either by time or by system. Cleared alarms and warnings are automatically removed from the alarms list upon operator acknowledgement. Both alarm and warning messages are displayed in fixed format consisting of the following fields:

- indication of alarm or warning;
- system in alarm;
- parameter;
- if an analogue scanpoint, present value, high and low limits, and units;
- if a contact scanpoint, present status;
- analogue or contact scanpoint which initiated the message;
- time of occurrence of alarm/warning.

Operator Monitor:

The Operator Monitor (Fig. 13) consists of a list of scanpoints which are selected by the watchkeeper at his discretion from the Propulsion Overview, the Alarms Overview and the various propulsion (engines and shafts), ancillary, and auxiliary systems pages. The entries in the Monitor list have the same content and format as messages in the Alarms Overview.

Secondary information requirements of the watchkeeper are provided through CRT pages for:

- engine start and drive control;
- fixed logs of engine parameters;
- engine data;
- shaft data;
- ancillary system monitoring and control;

- auxiliary system monitoring and control.

Engine Pages

There are a thirteen pages for monitoring and control of four cruise and main gas turbine engines. Associated with each engine are Control, Data and Log pages. There is also a log page for the two driving engines.

The Engine Control page (Fig. 14) displays the following:

- analogue and digital presentation of the four major gas turbine engine parameters;

- english-language messages giving engine status before, during, and after start, assume power and stop (or trip if the engine is automatically stopped due to alarm detection); and,

- elapsed time from instant of operator command input and during stages in the engine start sequence.

The status message series for a start to idle are as follows (note that each successive message overwrites the former);

(a) PERMISSIVES SATISFIED - ENGINE READY FOR START;

(b) HYDRAULIC START NORMAL;

(c) IGNITION ON;

(d) IGNITION SUCCESSFUL; and,

(e) IDLE SPEED ACHIEVED.

The Engine Data page (Fig. 15) displays the following;

- simple mimic outlines of both the gas generator and free turbine sections of the engine;

- nine gas generator scanpoints;

- six free turbine scanpoints; and,

- parameter type and data for each scanpoint.

The Engine Log page (Fig. 16) displays the following;

- current values of Ni, Nj, EGT and EGP plotted over the previous 30 minutes;

- alarm limit plots for each parameter; and,

- current values of all four parameters in digital format.

The Driving Engines Log page integrates the logs for the driving engines (two cruise or two main) on a single page.

Shaft Pages

The consoles provide two Shaft pages (Fig. 17), one for each of the port and starboard shafts. Each page contains a mimic diagram of the shaft, from clutches to propellers, and shows all scanpoint symbols and parameters together with labels identifying shaft sections and physical areas of the ship in which each shaft section is found.

System Pages

The Systems CRT pages for display at SHINMACS Machinery Control Consoles are process flow diagrams for each of the auxiliary and ancillary machinery systems. Most of these diagrams contain one or more control points, each of which can be selected via the Cursor Control module or, in certain cases, via line menus. Each diagram also displays all points scanned by SHINMACS and each scanpoint is selectable via the Cursor Control module for application of any combination of the three line menu-displayed information control options (i.e., DISPLAY ALL DATA; LOG POINT; ADD TO MONITOR). The main purpose underlying display of process flow diagrams is to assist the user (i.e., console operator) in modelling each system by visually presenting complete functional circuits in which machinery components and transmission of power, information, or material among them are logically and clearly represented.

Three examples of Systems pages are presented in order to illustrate page format and coding of alarms, scanpoints and machinery components (Figs. 18-20).

Machinery Command Input

All commands for control of machinery system components (except shaft knots commands in automatic propulsion control mode and engine power and propeller pitch in manual propulsion control mode) are made via the 24 keys on the bottom panels of the three CRT monitors. These keys are in line with the control command option line menus displayed at the bottom of the three screens. Operation of a key selects the command option directly above it on the associated screen. Most command option line menus are displayed in response to selection of control points on ancillary and auxiliary system pages.

Engine command option line menus are automatically displayed upon selection of engine control pages.

Control point selection is performed by moving the screen cursor via the trackball to the control point and operating a SELECT key.

Selection of:

- station-in-control (i.e., BRIDGE, LOP, or MCR);

LOP = Local Operating Positions
MCR = Machinery Control Room

or,

- propulsion control mode (i.e., AUTO or MANUAL);

is made via a line menu which can be displayed at the bottom of the centre screen. Automatic and manual control of gas turbine engines and propeller pitch is performed using four port shaft joysticks and four starboard shaft joysticks. In automatic propulsion mode, shaft RPM and propeller pitch are determined by demand schedules. The input to an active demand schedule is the knots control signal. Shaft RPM and propeller pitch are the output control signals of the demand schedule. Port and starboard knots are controlled directly by the two 'auto control' joysticks which can operate unlinked (i.e., independently) or linked. In linked mode, the knots control signals for both shafts are identical and can be controlled from either 'auto control' joystick.

Once the propulsion control mode is changed from AUTO to MANUAL, the auto joysticks serve only as telegraph devices. Manual control of gas turbine power and propeller pitch control signals is effected through three port joysticks and three starboard joysticks.

Supervisory Machinery Control Console (SMCC)

The SMCC is a single-operator, single-CRT workstation having controls for:

- line menu item selection;
- cursor control and control point selection;
- MMCC page monitoring;
- alarm acknowledgement;
- numeric data entry;
- voice communications;
- CRT monitor brightness and contrast.

All of the CRT pages available to the watchkeeper at the MMCC are also available to the watchsupervisor at the SMCC. In addition, pages are available for display of information from:

- scheduled, periodic machinery logs;
- demand logs;
- dynamic data analyses;

- health monitoring systems.

All CRT pages are selected via line menus displayed at the bottom of the CRT screen.

Line menus can also be used secondarily for backup control of:

- main propulsion machinery;
- ancillary and auxiliary machinery;
- propulsion control mode and station-in-control.

The Supervisory MCC is provided with several vertical and horizontal shelves for storage of operating manuals and printer output.

CONCLUSIONS

A human engineering design process has been applied to the development of machinery control consoles for SHINMACS based upon analyses of watchkeeper and watchsupervisor information and control requirements.

The design of SHINMACS machinery control consoles is based upon interactive graphics display systems technology which is considered to have many advantages over conventional process control instrumentation technology for information display.

This paper has described the major design features of SHINMACS machinery control consoles; namely, components, control functions, and displays and their operation.

The SHINMACS machinery control console design specifications developed by DCIEM have been integrated into the overall SHINMACS Statement of Operational Objectives. It is anticipated that a SHINMACS Service Test Module will be built and delivered to the Department of National Defence by 1983.

The Service Test Module will be delivered with a digital simulation of DDH-280 propulsion and auxiliary systems which will enable evaluation of SHINMACS design concepts, including those specific to the Main and Supervisory Machinery Control Consoles.

Console design concepts will be evaluated experimentally with trainee and fully trained SHINMACS operators performing monitoring and control tasks in the simulation environment, followed by similar evaluation at sea. It is anticipated that the results of these evaluations will be reported at the Seventh Symposium in 1984.

REFERENCES

(1) E.L. Gorrell, 'Human engineering considerations in design of information displays for machinery control consoles of future naval ships', Proc. of the Fifth Ship Control Systems Symposium, Vol. 4, David W. Taylor Naval Ship Research and Development Center, Annapolis, MD, USA, 1978.

(2) E.L. Gorrell, 'Human engineering design requirements for SHIN-MACS machinery control consoles: Part II. Console hardware and functions', DCIEM Technical Report No. 80-R-27 (Not For Release To Industry), Downsview, Ontario, Canada, 1980.

(3) E.L. Gorrell, 'Human engineering design requirements for SHIN-MACS machinery control consoles: Part III. Console displays and operation' DCIEM Technical Report No. 81-R-18 (Not For Release To Industry), Downsview, Ontario, Canada, 1981.

Fig. 1. Main and Supervisory SHINMACS Machinery Control Console Mockups.

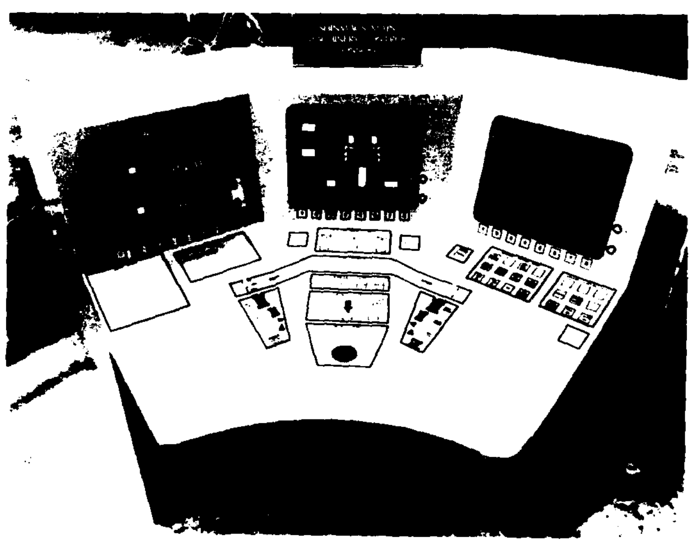

Fig. 2. Control and display layout for the SHINMACS Main Machinery Control Console.

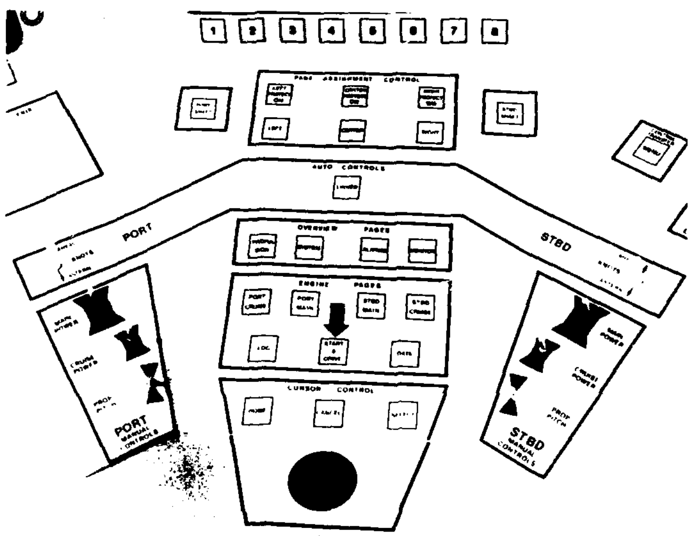

Fig. 3. Central controls layout for the SHINMACS Main Machinery Control Console.

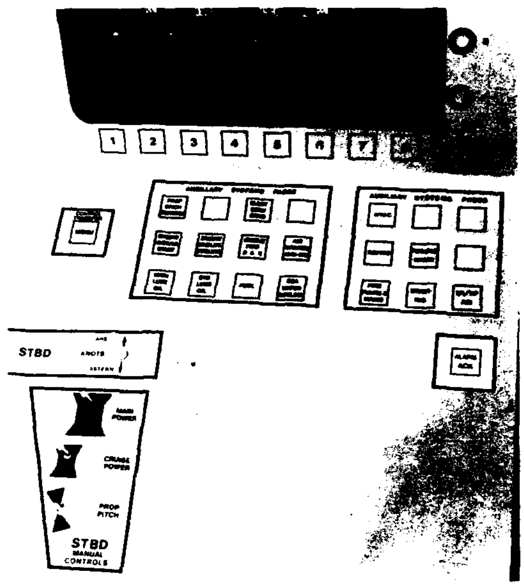

Fig. 4. Right panel control layout for the SHINMACS Main Machinery Control Console.

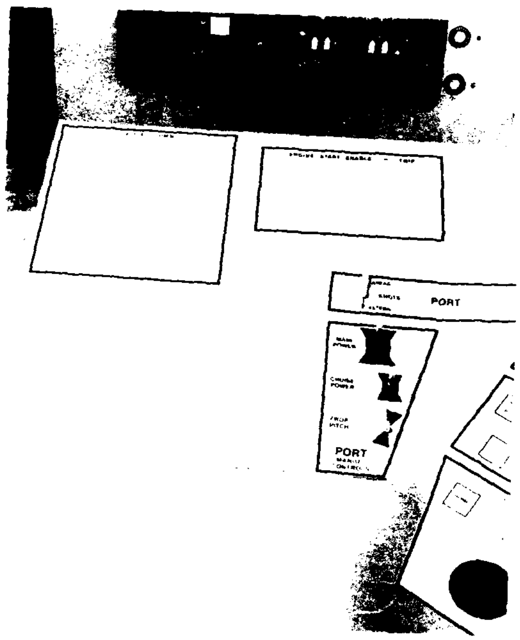

Fig. 5. Left panel control layout for the SHINMACS Main Machinery Control Console.

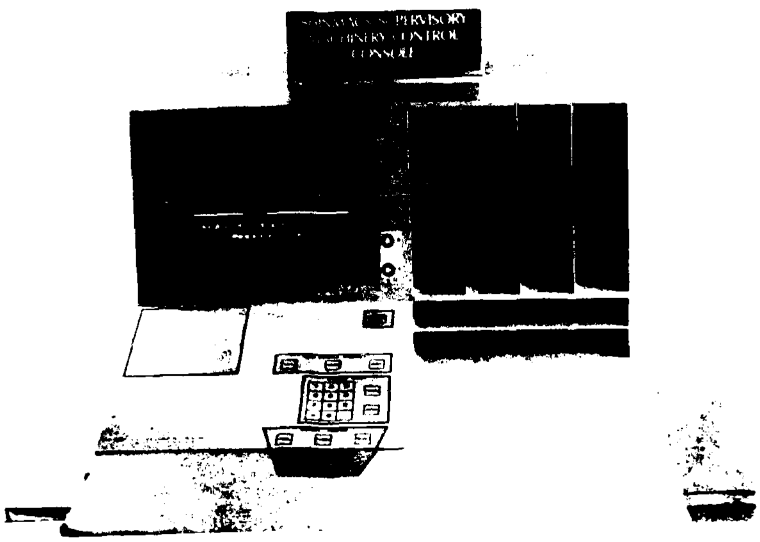

Fig. 6. Control and display layout for the SHINMACS Supervisory Machinery Control Console.

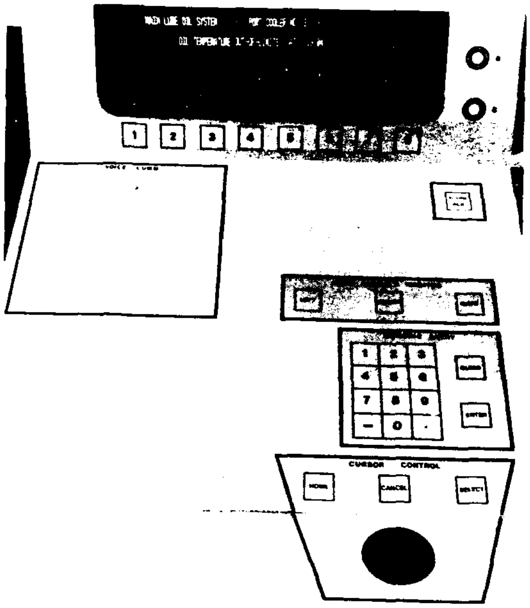

Fig. 7. Control layout for the SHINMACS
Supervisory Machinery Control Console.

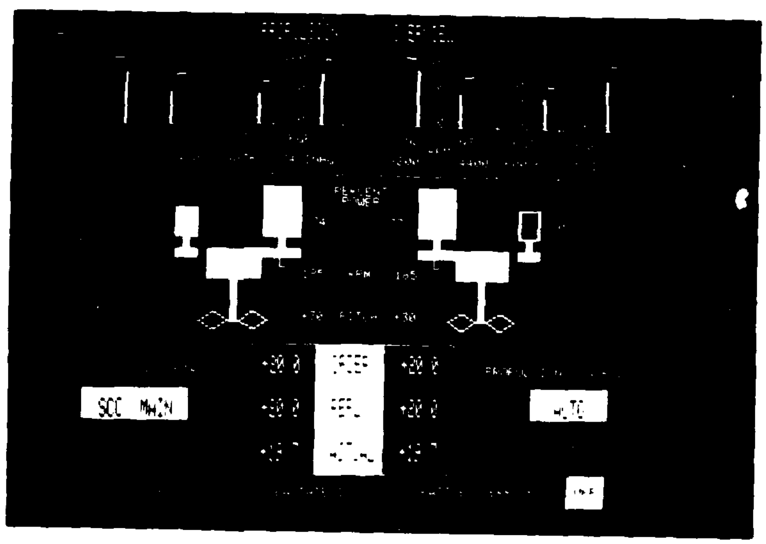

Fig. 8. Propulsion Overview CRT Face - SHINMACS Main MCC: Main propulsion machinery running normally with main gas turbine engines driving.

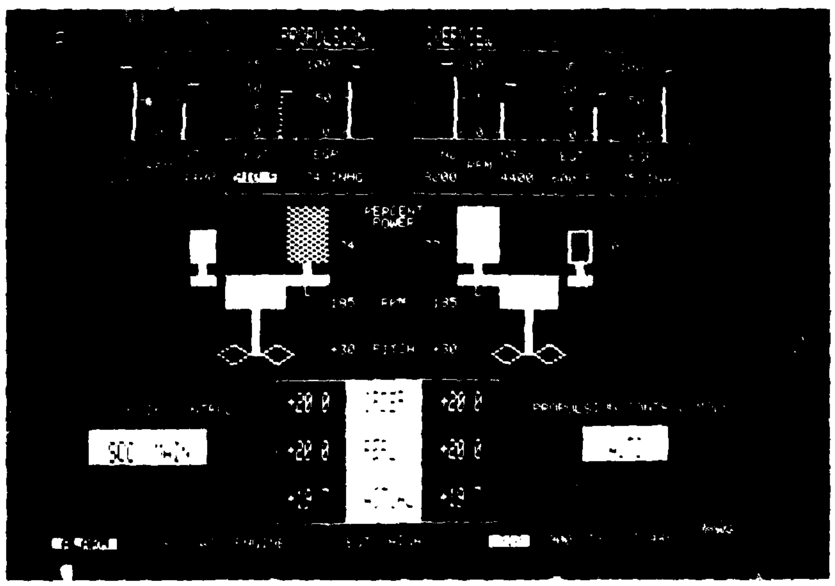

Fig. 9. Propulsion Overview CRT Page - SHINMACS Main MCC: Main gas turbine engines driving with an active alarm detected in the Port Engine System. The general message line shown in fi...

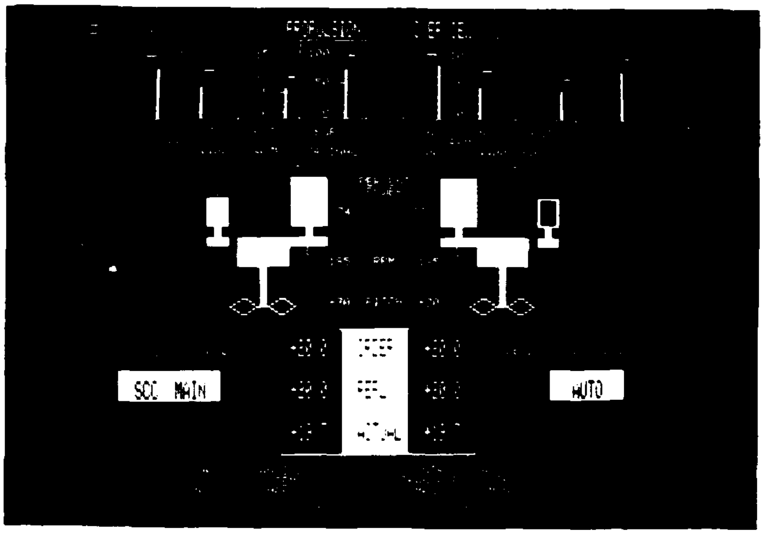

Fig. [illegible] ,n Overview CRT Page - SHINMACS Supervisory MCC: All CRT pages are accessed via the Master Menu shown in this photograph.

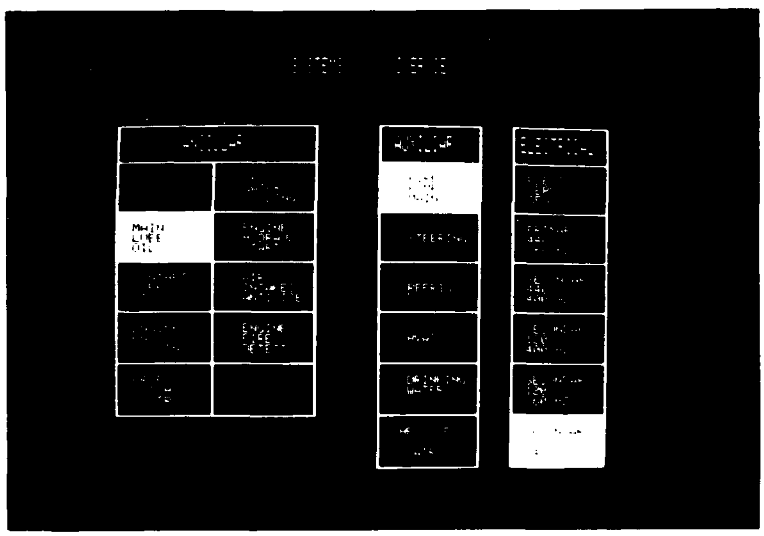

Fig. 11. Systems Overview CRT Page - SHINMACS Main MCC: Active alarms are shown in inverse video for three different systems.

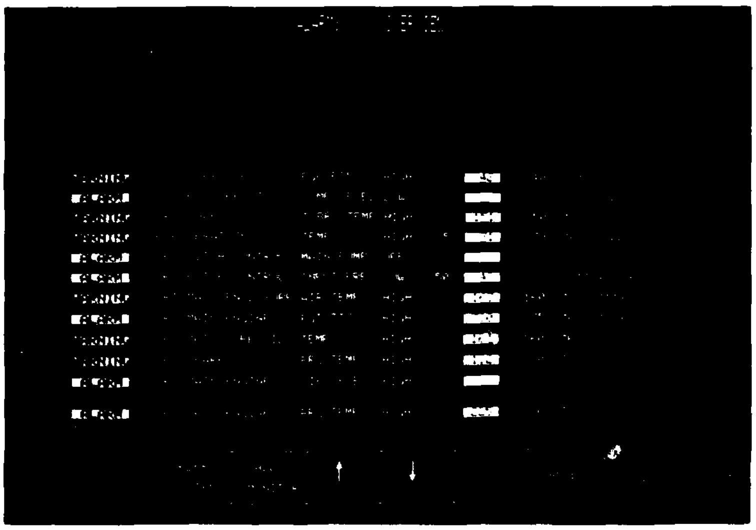

Fig. 12. Alarms Overview CRT Page - SHINMACS Main MCC.

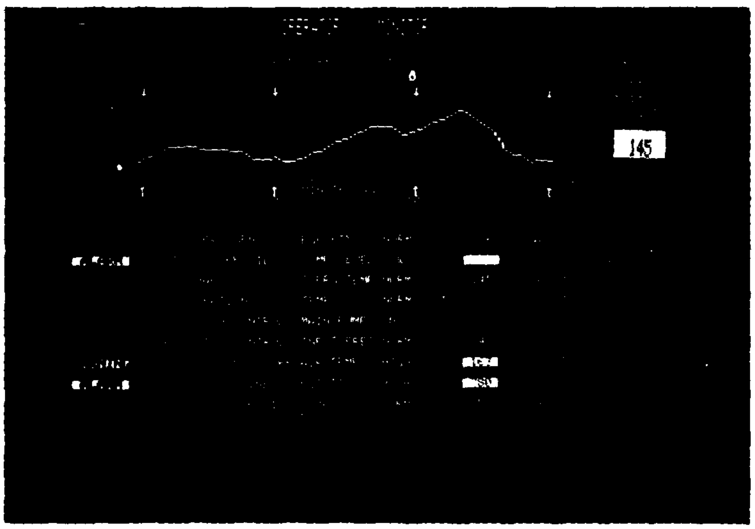

Fig. 13. Operator Monitor C.:T Page - SHINMACS Main MCC.

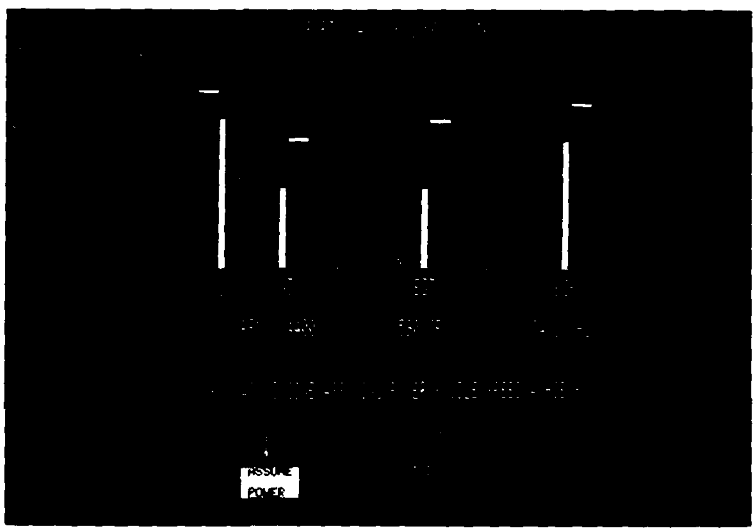

Fig. 14. Engine Control CRT Page - SHINMACS Main MCC.

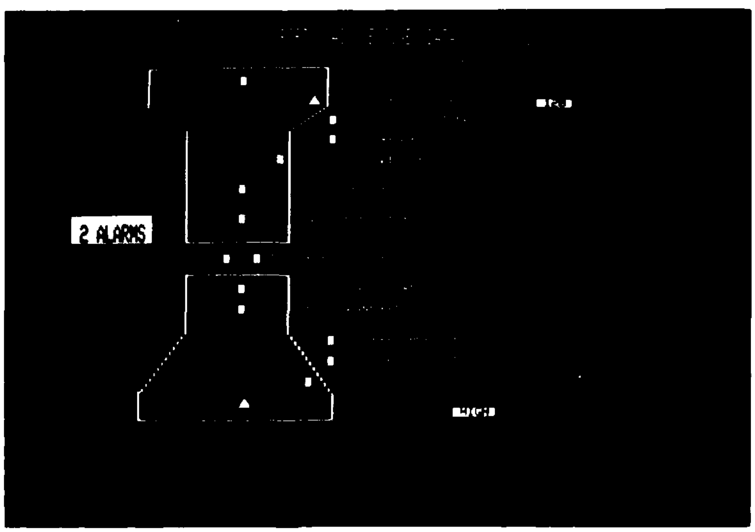

Fig. 15. Engine Data CRT Page - SHINMACS Main MCC.

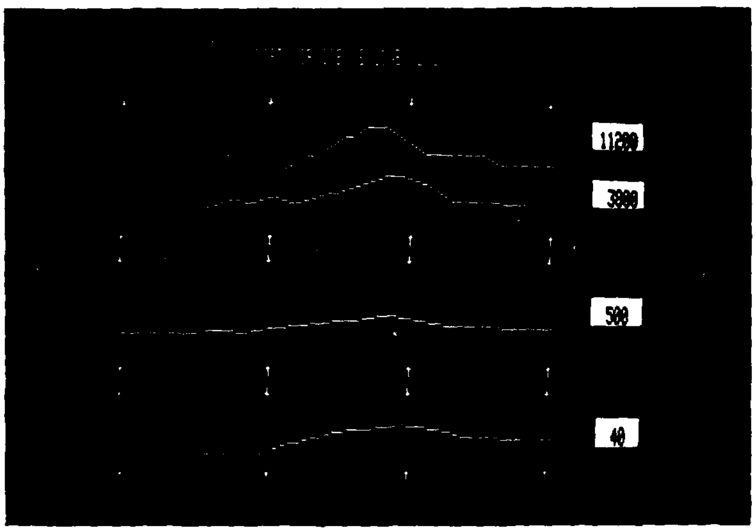

Fig. 16. Engine Log CRT Page - SHINMACS Main MCC.

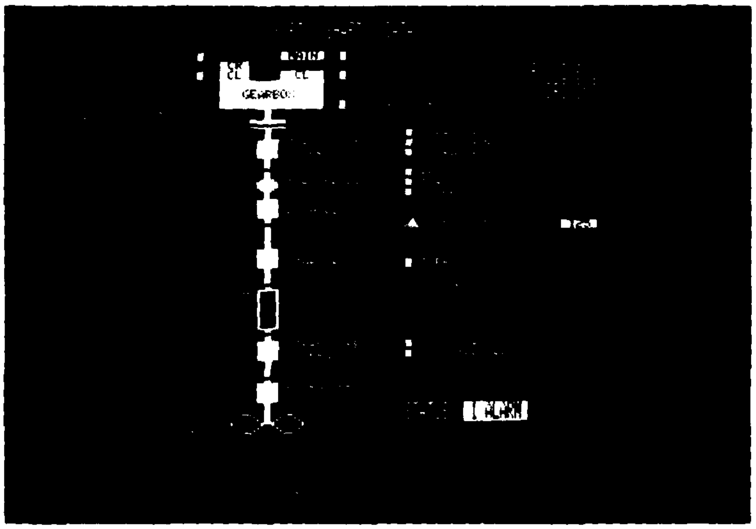

Fig. 17. Shaft Data CRT Page - SHINMACS Main MCC.

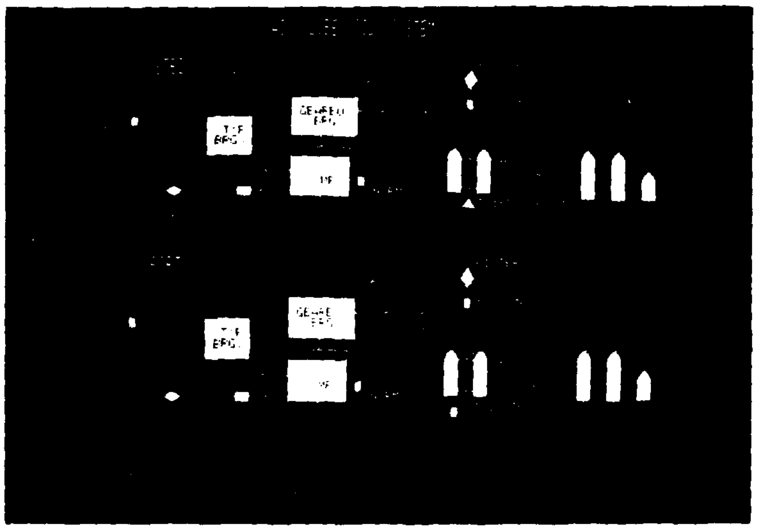

Fig. 18. Process Flow Graphic CRT Page for the Main Lube Oil Ancillary Propulsion System - SHINMACS Main MCC.*

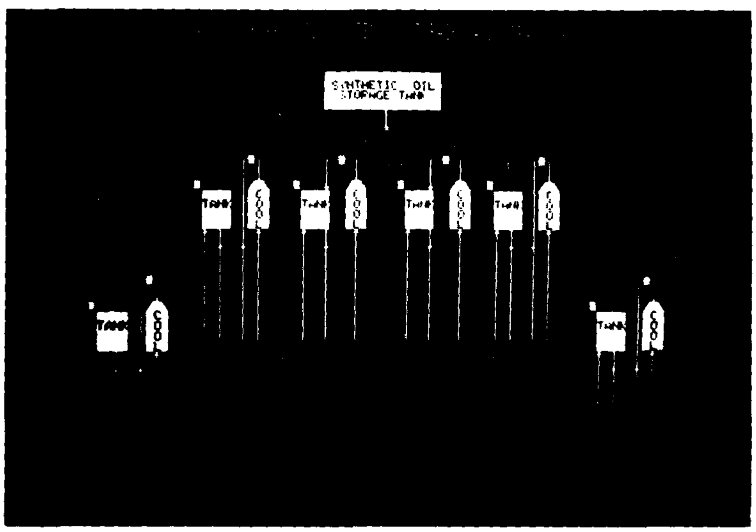

Fig. 19. Process Flow Graphic CRT Page for the Storage and Transfer Subsystem of the Synthetic Lube Oil Ancillary Propulsion System - SHINMACS Main MCC.

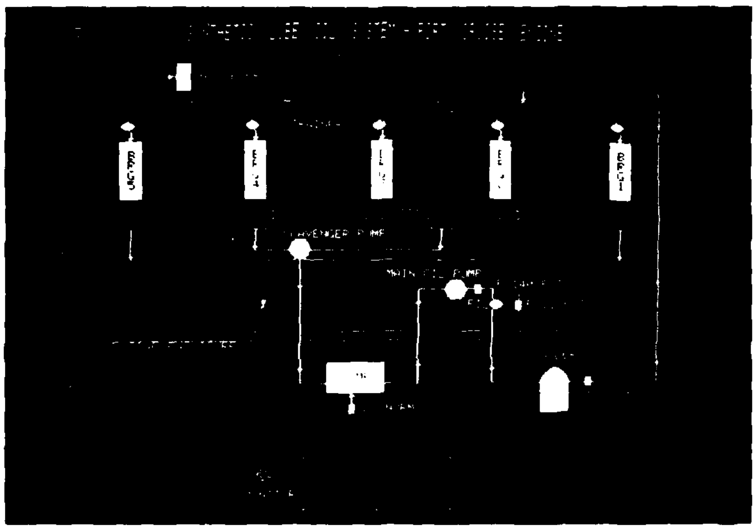

Fig. 20. Process Flow Graphic CRT Page for the Port Cruise Engine Subsystem of the Synthetic Lube Oil Ancillary Propulsion System - SHINMACS Main MCC.

AN ADVANCED CONCEPT IN INTEGRATED SHIP CONTROL SYSTEM DESIGN UTILIZING DISTRIBUTED MICROPROCESSORS AND STATE OF THE ART MODULES

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ABSTRACT

The ship control system designer has, in the last few years, been relieved of the drudgery of numerous detailed circuit designs, repetitive calculations, and module development. This type of detail design has not been limited to circuit design, but has extended into systems development of circuit modules, single chip digital computers, single chip analog computers, circuit simulators, high level software language development tools, and systems network development. The control system designer is left with a high degree of freedom at a lower risk and lower overall cost.

The integrated system concept involved today embodies multi-technical disciplines of system engineering, design, hardware engineering, design, hardware engineering, mechanical, reliability, maintainability, cost, manufacturing and training. Other considerations are packaging, human factors engineering, operability, and functionality. This paper will discuss the total design concept and multi-disciplined approach taken in the design of a new generation ship control system.

The advanced concept design utilizes numerous technologies to achieve low cost, high reliability, low weight, high degree of integration, low maintenance, easy repair, operator interaction, low risk, and a high degree of commonality. The design criteria was kept simple; design a flexible control system to ship equipment interface, develop simple man-machine interface, and maintain high degree of flexibility in the design modules. This philosophy resulted in the application of the following types of technologies:

Laser/fiber optic cables and data transmission
Multi-processing and distributed controls, both final digial and analog designs
LED lamp annunciators and serial communications
PLASMA & Cathode Ray Tube interactive displays
Multi-redundant data transmission techniques
Networking, packet switching, protocol and data security
Optically coupled input receivers and output drivers
Matrixed and terminal type keypads and keyboards
Flat cables and solderless connectors
Combined software and hardware simulators/emulators
Software development stations
Combined High Level Language and Assembly Language programming

This paper will discuss the trade-offs and studies used in arriving at the decision to apply the foregoing, and how each is app ed to meet the specific goals set forth.

Present System Design

Existing control systems designed in the seventies utilized the then current 'state of the art' design techniques for Control Systems and numerous data acquisition functions. The design called upon Mil-Spec parts where possible, and wrote special procurements for unavailable parts. The design has been the topic of numerous papers on Control System design since its inception and operation in service. Numerous papers have discussed the Control System, the propulsion system and the control philosophies, along with its benefits, advantages and disadvantages. (Ref (1))

The design of the Control System was based on much of the traditional Navy watch standing and logging philosophies. Hourly logs that were kept by hand were kept by automatic logging. This logging of hourly data is unnecessary, and costly in time spent evaluating the data and the cost of paper. The hierarchy of controls from the bridge, to the Engineering Operating Station (EOS), to the local operating stations (LOS) was standard. The fail safe and fail soft philosophies were set to carryout man intervention. The circuits for various subsystem control functions were partitioned so as to provide functional independence. This hierarchy of functional independence necessitated numerous cards to be utilized in a series fashion such that 5 to 7 circuit cards were used to process discrete, analog, logic, signal conditioning and output discretes.

The serial data bus design used in earlier Control Systems were a synchronous time division multiplex data bus utilizing data, clock, sync and con-rol lines. This design was the 'state of the art' and provided simplistic error detection schemes of clocks or loss of periodic synchronization. The cost of minicomputers, at the time this design was being implemented, was prohibitively high to use in multiple processing or control. Multiplexing of data was done with discrete or small scale integrated circuits. Numerous control cards made up multiplex data lines. This scheme, although complex and seemingly expensive, saved considerable wiring and numerous discrete signal conditioning cards. The system utilized triple redundant cables and a loop communication data. All of the data was clocked and put in time slots.

The available small scale integrated circuits (SSI) and medium scale integrated circuits (MSI) were primarily of the high power variety in 1970, with the advent of Low power (L), Schottky (S) and Low power Schottky parts (LS) becoming available by 1975. Hence, the integrated circuits (IC's) of 1970 variety consumed as much as 20 times more power than the 'LS' parts of today. The design utilized forced air colling over the cards to remove the heat. These circuits were quite 'noise' prone to line spikes or voltage fluctuations.

The electric readout meters considered of Mil-Spec arsonal movement meters that are sensitive to magnetic interference, shock and vibration, and required periodic calibration. The meters were large and quite heavy, although they were easy to read.

The overall system design of a typical 1970 technology Control Ssystem consisted of about 1100 circuit cards and 29 console sections. The weight of the total consoles was 21,650 pounds. The number of indicator lamps on each console is so great that lamp tests are conducted by console sections rather than by the console as a whole.

Technicological Advances

Since 1970, technological advances have been made in logic circuits from SSI and MSI circuits to large scale integrated circuits (LSI) and ultra large scale integration circuits (ULSI). Designs that once took 3 circuit boards to implement is now done with one LSI device. The microprocessor advances since 1975 from 4 bit, to 8 bit, to 15 bit and now to a 32 bit micro main-frame computer design staggers the imagination. Single chips now perform frequency synthesis, speech synthesis and recognition, power conversions, analog to digital (A/D) and digital to analog conversions (D/A). There is now an analog computer on a chip that allows 4 real time inputs with 8 real time analog outputs. The device can perform complex filter operations extending to 20 complex poles and 20 zeros.

Development in commercial communications has enabled fiber optic light emitting didoes (LED's) and solid state LASER's to be developed with a highly controlled frequency specturm to enable up to 100 million bits of data per second to be transmitted over fiber links. Advances in fiber links, glass and plastics have allowed the fiber to be strong, flexible and light. Several hundered feet of fiber and sheathing weighs only a few pounds.

The advances were not only limited to hardware development, but grew to include system design development. The available software tools were advanced beyond assembly language and the normal high level languages, but have included analytical tools for model stimulations and complex filter designs. Advances in filter design are brough about by the advances in technology and consequently filter design can be made on line and sample cases tried out through in circuit emulation or software simulation. With satisfactory model data and simulation, the model can be converted into a machine code and a prom generated for hardware simulation.

Design tools for digital design now included in circuit emulation for numerous processors from one development station, so that the investment on development hardware is minimized. The software design tools for debugging, and high level language programming have greatly increased the programmers productivity. It is now shown that approximately 80% of design costs today are allocated to software development as compared to 10%, 10 years ago. The hardware costs are drastically lower due to great advances in microcircuit development. The biggest investment in the state of the art design today is in the software station and development software operating systems. High level languages such as PL/I, PL/M, PASCAL and ADA offer machine transportability, but represents a large investment in developing the language for the targe machines.

System Goals

Keep the interface between man and machinery as common as possible. Keep the same input and output control functions and maintain the itegrity of the critical controls. Maintain a high degree of commanlity, low cost, and be maintainable by ships personnel with minimal training.

System Goals

Reduce weight where possible, reduce cabling and power. Along with power reduction there is reduced coding requirements.

We tried to apply proven technologies where the fit of technology to the application is good and still maintain the simplicity and reliability of the other design.

Trade-off Studies

For our trade-off studies we evaluated future ship control system requirements, and tried to come up with a design using present technology to meet two requirements in the most cost effective manner.

Subsystem Controller. Our immediate tasks were to identify all the numerous support circuit cards that performed in series to make a machinery subsystem part of the automated system. The first subsystems brought to mind were the fuel oil service, lube oil service and sea water service systems. These subsystem utilized two electric pumps of varying speeds and placed one in operation while the other was in standby. We found that discrete signal cards, analog signal conditioning cards, analog to digital cards, comparators, output drivers and logic circuits were required. These cards were quite common and repetitive in design. They all have basically the same number of inputs and outputs. Only the logic and timing changed between systems. The requirement could easily be filled with a programmable controller with input and output signals. The cards would then be common except for a program change. We set the circuit card on a larger format to accommodate a higher pinout capacity and IC area. We utilized one card to house a single chip multiplexed A/D with a single chip microcomputer and ultra violet light reprogrammable read only memory (UVPROM). The input and outputs are buffered. (See Fig. 1), and a communications link is provided to communicate data to a host processor.

This card formed the basis of a distributed microprocessor designed controller. With further investigation we designed a method that enables a "watchdog" type circuit card to enable and disable the controller card that in the event of a failure on the bus, the failed card can be isolated. Further isolation is provided via the control override down stream of this card at the driver levels. The card is set up such that pulling the circuit card will cause it to fail soft and in full manual control.

We imposed self check and diagnostic routines on the card so as to provide remote automatic testing while in line and ease in diagnosting circuit problems. Since each subsystem can be unique in its control, the program and self checking logic enables testing of the status 'key' word contained in a micro-switch to prevent inappropriate cards from operating in the wrong slots and also to key the on-board programming system that programs the card for one of the multiple uses.

Microprocessor Selection. There are 8 bit, 16 bit, and 32 bit microprocessors commercially available that can be used for multiprocessing. Here, the key to selection is not only computing power, but the availability of support software for the micro-

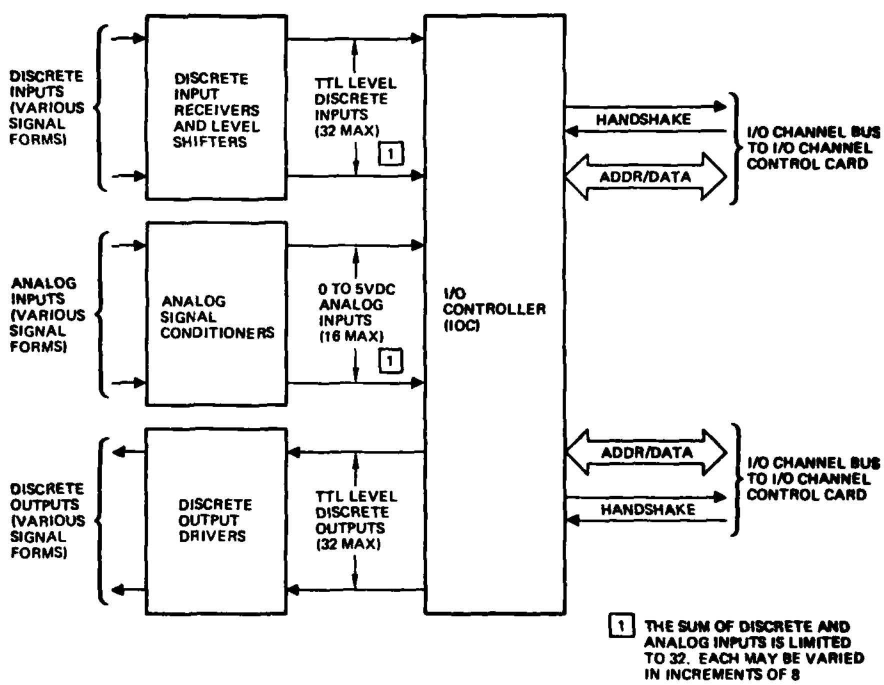

FIGURE 1 GENERAL PURPOSE SUBSYSTEM CONTROLLER

processor. The newest 32 bit micro mainframe processor, although fast and commercially available, it is still too new and untried to put into a military design. There are numerous 16 bit processors on the market many of which profess military temperature and environmental specifications, although without the MS-38510 dash number yet assigned.

We evaluated the numerous processors available for hardware capabilities, support devices as DMA, arbitration, I/O processing and numeric processing ability. Next we evaluated the availability of software tools and development software high level language availability and available programmers. Software support and availability holds the key to the cost effective design since software costs now contribute 60 - 80% of the budgeted cost of new engineering programs. It once only contributed 10 - 20%, but with the advent of low cost and powerful micro-computers and large scale integrated electronics, this position has changed drastically. We further, needed assurances that the processor support software is timely and fully supports the hardware availability, and not lagging it. Operating systems, real time scheduling and interrupts are only part of the software support requirements.

Lamp Indicators. Next, we evaluated the lamp indicators and the schemes for indication. The lamps were primarily used to indicate status or alarms. For status indicators, it is easy, the normally isolated contact is powered with the same current and tied directly to the indicator to indicate open/close and on/off, etc. For alarms, it is a different story, there are several circuit cards involved in for signal conditioning, set point comparison, alarm time delays, hysteresis, flasier, audible alarm drivers and lamp drivers. Due to the circuit partitioning requirements and functional isolation may extra cards were used to provide isolation such that removal of a card did not cause another subsystem to fail to operate. In this case 5 separate circuit cards were involved, signal conditioning, alarm set point, time delay, flash control and output drive to switch 28 volt lamp driver transistors.

Microprocessor technology with electrically alterable read only memories (EAROM's) provided the set point comparison, time delay, hysteresis, flash and alarm control logic. The combination of 2 types of circuits and processing provided the condition signal and the response lamp and flash. The lamps were changed to LED light bar designs which have several hundred thousand hours MTBF, and draw 10 ma at 5 volts. The lamps can be configured with 2, 4, or 8 lamps for brightness and reliability. Training of personnel, along with maintaining skill levels of previously trained personnel is becoming a problem in the armed forces today. The liberal use of mimics for control and display eased some of the burdens, but still, the size of the control consoles became prohibitively large and formidable to the operators.

Human factor engineers are trying to cope with the problem by offering compact logically arranged computer driven displays as Cathode Ray tubes (CRT's) and plasmas panel displays. Color versus monochrome is another topic of discussion, but with inverse video and other highlighted enhancements, monochrome displays are used to draw attention and quickly present to the operator pertinent data. We considered color, plasma and monochrome CRT for application, cost, safety and practicality. Color is highly desirable, but the most expensive. We concluded that with the size and weight constraints as well as costs, a mix of 3 plasma and 1 CFT display utilized on the console and supervisor's display respectively is the best match. The CRT would offer the high resolution and rapid response while the plasma offers lower cost, space and weight at the console levels.

Cable Design. Cable assembly trade-offs were completed to determine the most cost effective way to wire front panels and circuit card cages. Cable connections are presently made by soldering and/or crimping connections. Flat cables and locking connection assemblies change the assembly time to cutting and crimping the cables without the need for dressing the cables. This necessitates several changes to panel assemblies, such as pushbuttons now become grouped with connectors and lamp displays are grouped rather than individually wired.

The New Design

The newly configured system design selections were made following the extensive research and evaluation of the modules, costs, reliability and technical risks associated. The modular approach required fewer design hours to evaluate since larger scaled integration covered most of the design interface problems of going from one device to another. By utilizing families of circuit components that are by manufacturers design, fully compatible in timing, controls and electrical interfacing, circuit and component selections were simplified. The application of standard protocol and busing techniques again, minimize the interface difficulties. The selection of software development tools and modules that are family compatible, and utilizing higher level languages offer ease of integrating software and hardware, by relieving the programmer of designing operating systems, assemblers or compilers, but to allow him the latitude in writing program and executive software.

Savings. The new design reduced the circuit cards to approximately 400 circuit cards. The number of unique card types are approximately 45 types. Power dissipation is approximately 1650 watts, which eliminates the need for forced air cooling fans. The console sections goes from 29 sections and approximately 21,000 pounds to 15 sections and 9.450 pounds. The estimate of cable weight savings is 20 tons due to higher degree of multiplexing and glass fiber cable over wired multiplex systems.

Subsystem Controller. A subsystem controller card utilizing a microprocessor, 16 channel multiplexer and A/D (one chip), buffer and drivers is used in replacing approximately 5 circuit cards for each of the pump control subsystems as fuel oil systems, lube oil system, CRP pump control, sea water pump control and steering pump control logic. The new design uses one card with separate program and key code for the various applications, and replacement circuit boards are programmed as needed aboard ship. The number of spares is reduced owing to the commonality of the card replacement, and the programmable design. Of course, future updates in the design and changes to the logic are only software changes.

Serial Data Transmission. The selection of fiber optic data transmission enables higher data rates with lower probability of error due to RF or EM interference. Fiber transmission also allows larger weight savings, which is reflected in eventual fuel savings to the ship. The fiber transmission further allows secure data transmission and eliminates common mode and ground loop problems common to wired systems. The new design consists of 3 sets of transmitter and receiver cards at each station enabling direct communication with built in protocol, error checking and retransmission. (Fig 2) The intelleigent controllers for data transmissions have opened up a realm of data communication through microprocessors that data packets or messages are automatically retransmitted or re-routed to the correct receiving station. Here, the communications controller device is so sophisticated, the network design engineer had to decide the best device based on cost, speed and availability, rather than design and implement all the complex correction and checking circuits himself.

Display Systems. Interactive displays were included in the combined electric plant and propulsion control console. The displays are plasma type owing to the flatter configuration and lower voltage at the front panels. The displays will provide mimic like displays and with enhanced video to show faults or faulty areas. 'Page-type' formats enable rapid scan and data presentation.

A unique alarm indicating module design utilizing a single chip microcomputer, serial interface and LED buffer and drivers form a part of the status and alarm indication system. (See Figures 3, 4, and 5). The module consists of 2 cards, one is the LED light bar group and second the receiver and driver electronics. The modules are identical throughout the ship and only the transparent lengen overlay changes. The central computer transmits to the modules several data words for flash, steady or test mode. The receiver puts the data words in storage, and a run program executes the flash mode for the lamp identification line or steady as required. During test, the self contained microcomputer executes a pattern on the display and also provides a test word to the central sys tem to check for validity. In this fashion the alarms can be changed by simply identifying a new position to be illuminated from the spare and make the software change in the central system. The lamps are further backed up by discrete enable lines brought on to the card as required and also by one of the 3 plasma inter-active displays.

System Comparisons

The size and weight comparison between the earlier design and the new design is shown in Table 1. This does not account for the additonal ship weight in cables or the battery for the uninterruptable power supply.

The card comparisons for the earlier design is shown in Table 2.

We selected fiber optic and laser transmitters for high data integrating, low cost, low weight and high noise immunity. The selection and applications of HDLC devices over the old design serial data has provided high data rates and intelligent control for automatic retransmission, error detection and correction.

The selection of a multiprocessing system and distributed controls provides the flexibility of microprocessor designed circuits and a high commonality of circuit boards. Selection of a family of processors and interface devices eased the burden of detailed timing analysis circuit interface designs and arbitration design. Single

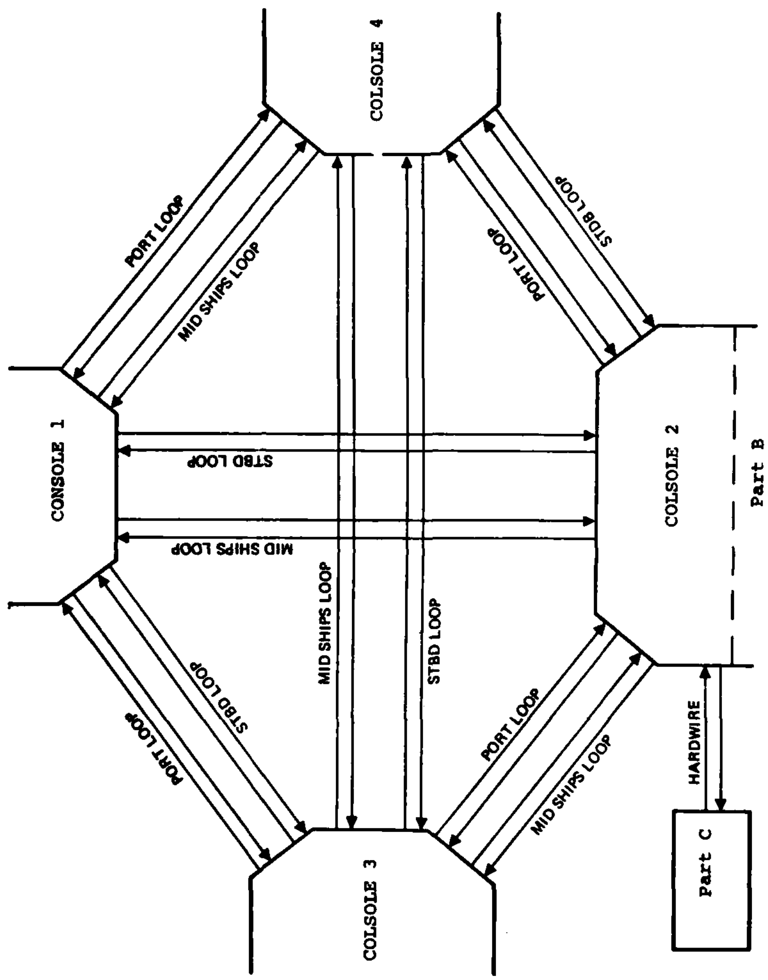

FIGURE 2 TOTAL FIBER OPTICS NETWORK

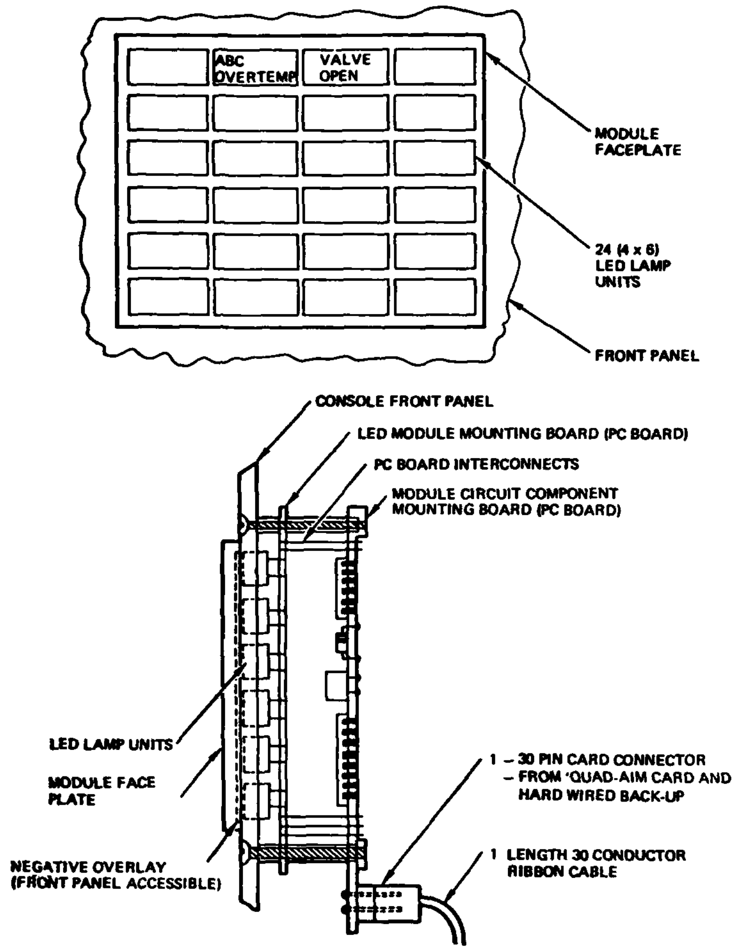

FIGURE 3 ALARM INDICATING MODULE, LAYOUT & CONFIGURATION

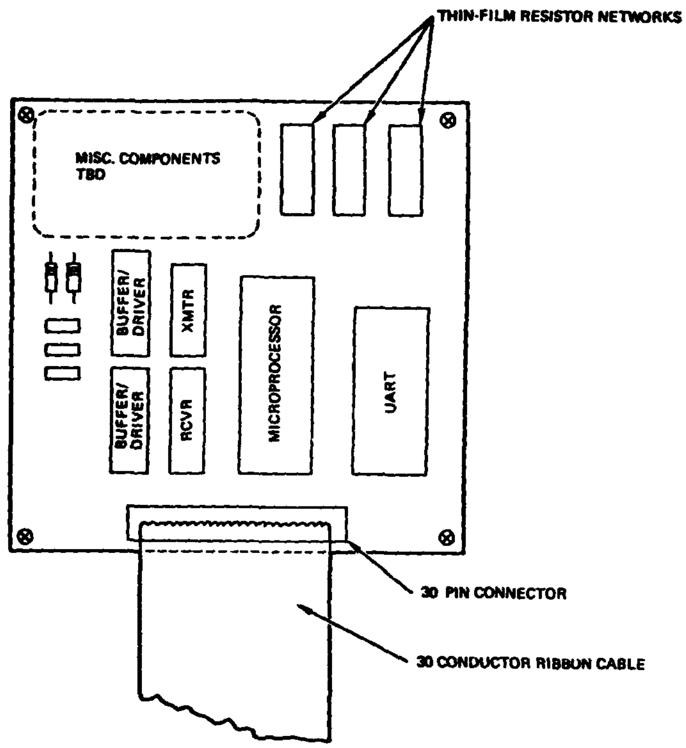

FIGURE 4 AIM CIRCUIT BOARD CONFIGURATION

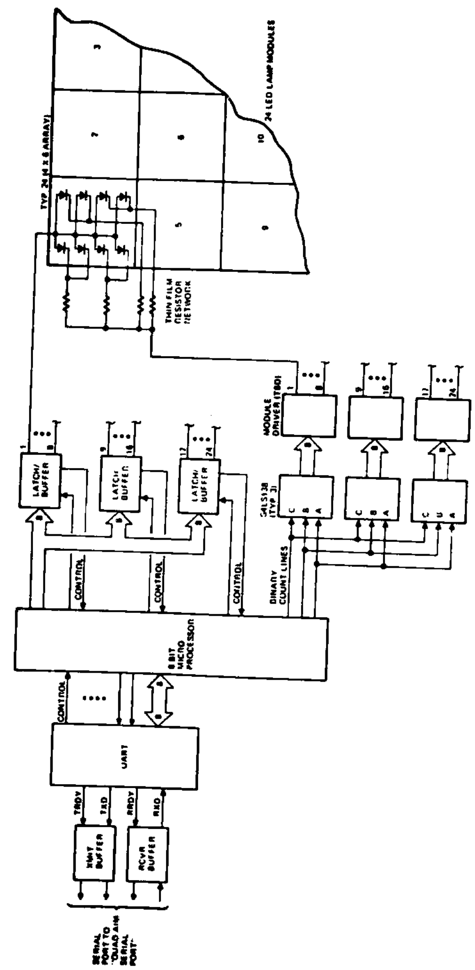

FIGURE 5 AIM CIRCUIT BLOCK DIAGRAM

TABLE 1 SIZE AND WEIGHT COMPARISON OF AMICS VERSUS EXISTING SYSTEM

Existing System			AMICS		
Console	No. of Bays	Weight (Lbs)	Console	No. of Bays	Weight (Lbs)
CONSOLE 2	5	3148	CONSOLE 2	3	2000
CONSOLE 5	4	3753			
			CON2 (New)	1	600
CONSOLE 2	3	1850	CONSOLE 2	2	1250
CONSOLE 3, 4 (2 Units)	6 (3 each)	4418	CONSOLE 3, 4 (2 Units)	6 (3 Each)	4420
CONSOLE 1	3	1161	CONSOLE 1	3	1180
System Power Supplies	8	7325			
Total Per System	29	21,655	Total Per System	15	9,450

TABLE 2 COMPARISON OF CARD COMPLEMENT

Existing System			AMICS		
Console	No. of Cards	No. of Modules	Console	No. of Cards	No. of Modules
CONSOLE 2	227	0	CONSOLE 2	59	37
CONSOLE 5	381	0			
			CONSOLE 3	6	2
CONSOLE 3	65	0	CONSOLE 3	37	7
CONSOLE 4,5 (2 Units)	380 (190 each)	34 (17 each)	CONSOLE 4,5 (2 Units)	174 (87 each)	54 (27 each)
CONSOLE 1	28	1	CONSOLE 1	20	10
Total for All Consoles	1081	35	Total for All Consoles	296	110

device bus arbitration devices provide DMA and bus contention control. By utilizing a dual bus concept, we have provided higher reliability in the design, so that one bus can cause the system to fail. Multi-processing provides data and control integrity by 'voting down' a bad processor, and enabling the built in diagnostic routines. Distributed processing relieves the central control of mundane tasks and provides the system designer with better control over subsystems by enabling the design to more closely follow the traditional logic control and interface design. The addition of a bus interface and 'watchdog' circuit to the subsystem (distributed controls) provides fault isolation and detection to operate in the background to all vital tasks, and with full safety of fail safe features.

The LED lamp annuciator and serial communication design offers lower cost per lamp indicator and extremely high reliability owing to the LED reliability and back up hardware and support plasma displays. The LED alarm indication module demonstrates the design flexibility of add on or removal of alarms by software table changes. The built-in communication for test and self testing will automactically set a precidence and standard for alarm annunciation. The LED lower power and high light dispersion assimilates the lambertian light distribution of conventional incandescent lamps. Replacements can be made easily without wiring changes or program rerouting.

The selection of Plasma and CRT technology combines the space and weight profiles to make the design compact, pleasing to the operator and video display capabilities.

We selected redundant data links, but utilized single chip controllers and protocol devices to handle the data transmission and integrity. The fiber optically coupled data with 3 paths to each mode and 6 paths with message switching provides highly reliable transmission of data.

We utilized optical isolated receivers and drivers to minimize loop problems assocaited with transistor coupled level shifters. Hence, the optical isolation provides ground noise immunity and relieves the high voltage coupling on board.

The appliation of flat cables, solderless connectors and matrixed key-boards and keypads provided large cost savings in assemblyl calbe harnessing and panel size and weight.

Conclusions

The results of this case study are not final yet, as the hardware and software has yet to be built. The state of key breadboard and software development has us excited about achieving 40 - 50% savings in cost, weight and power. We see that the future of this type of design enhances the system reliability at lower costs, and provides training and diagnostic maintenance capabilities never before used in the Naval Ship Control design.

REFERENCES

(1) Proceedings Fifth Ship Control Systems Symposium, October 30 - November 3, 1978, "An Evaluation of Simplified Propulsion Controls For Gas Turbine Powered Naval Ships", D. A. Rains, and I. Friedman.

REMOTE DATA TELEMETRY IN A SHIPBOARD ENVIRONMENT

by Albert J. Van Vrancken
TANO Corporation

ABSTRACT

The commercial marine industry is long overdue for a change in attitude toward the application of state-of-the-art technology to marine automation. The petrochemical industry has applied computer-driven supervisory control and data acquisition systems to its information needs since the early 70's. U. S. built shipboard control systems have undergone very few evolutionary changes in operational philosophy since the late 60's.

This paper presents some concepts in marine control system architecture which contribute significantly to cost reduction, safety, and modernization of the marine industry. The application of microprocessors, color video displays, remote data telemetry, and quarters' area control to the marine propulsion plant is detailed in this paper.

INTRODUCTION

TANO Corporation has been designing and manufacturing automation and control systems for the marine and petrochemical industries for more than 10 years. Our experience gives us some license in comparing and contrasting the technological advancement of these two industries.

THE PIPELINE INDUSTRY

For over a decade, the oil and gas industry has been using supervisory control and data acquisition (SCADA) systems to gather information from and to control complex pipeline networks. These webs of pipe carry virtually any type of fossil fuel, including crude oil, refined products, and natural gas. Similar systems are used for on-site operation of refineries, LNG plants, and even waste water treatment facilities. The common denominator for control and monitoring of these critical activities is serialized data transmission.

Faced with the need to control pipelines hundreds of miles long, the petrochemical industry utilized remote data telemetry as the most viable means to accomplish this task. Using telecommunications technology, a pipeline dispatcher can open a valve and start a pump five hundred miles away with a few keystrokes at a cathode-ray tube (CRT) terminal.

SCADA: How It Works

The typical SCADA system (See Figure 1) usually consists of a Master Terminal Unit (MTU) and a number of Remote Telemetry Units (RTUs) scattered along a pipeline at compressor stations, injection sites, and delivery terminals. The MTU is generally comprised of a host Central Processing Unit (CPU), mass storage device, hard copy logger, communications interface and a CRT/Keyboard, which functions as the primary man/machine interface. The RTUs are capable of accepting and issuing both digital and analog signals. Digital inputs consists of binary or contact-type signals indicating the state of a valve or pump, alarm contact, pressure switch, etc. This classification also includes pulsing signals, such as those which might come from a flow meter. Analog input information is in the form of current, voltage, RTDs and thermocouples. Digital output takes the form of relay contacts for use in valve and pump operations. The analog counterpart provides the setpoint for control of varying process parameters (ie, flow control, temperature variance, pressure control).

The sequential polling of the RTUs by the MTU provides the operator with real-time information regarding pipeline events and parameters. When a command is entered at the CRT/Keyboard, this scheme is interrupted, and the command is immediately transmitted to the RTU at which the control point is located. Polling then resumes at whatever point it was interrupted.

THE SHIPBUILDING INDUSTRY

In contrast, the typical U. S.-built commercial marine automation system is comprised of one or more large consoles densely packed with indicators, pushbuttons, meters, gauges, printed circuit cards, relays, and miles of wire. This maze of electronics and pneumatics is then connected to the propulsion plant and its auxiliary systems via additional miles of cable and tubing supported on tiers of wireways and pipe hangers.

Two Eras of Technology

As a result of an apparent conservative attitude by owners, shipyards, and regulatory bodies, the commercial marine industry is at least 10 years behind the pipeline industry in the application of state-of-the-art technology to automation.

In 1971, TANO Corporation delivered its first computer-driven supervisory and control system. This system was preceded by several years of hardwired schemes which still involved remote telemetry. In July, 1981, the first commercial marine system having some form of intelligence incorporated into the monitoring system was completed at the TANO facilities and delivered to the customer. However,the processors were used only for alarming and monitoring of the non-regulatory body required functions. No control was implemented via the CPUs. It should be noted that foreign-built ships have been using computer technology in marine automation for several years.

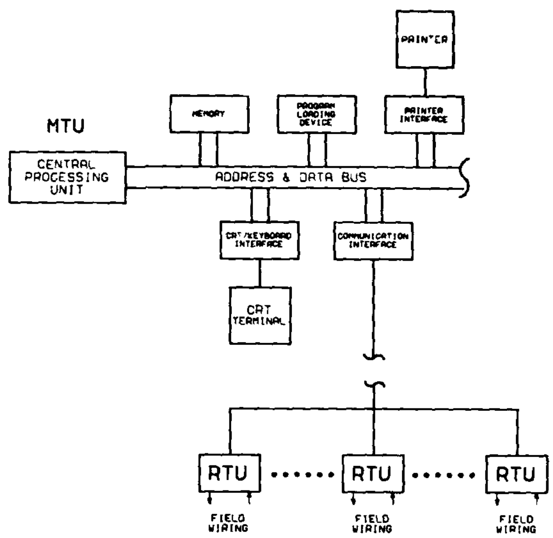

FIGURE 1. TYPICAL SCADA SYSTEM

Maintenance

The level of expertise of typical shipboard operating personnel, coupled with short tours of duty, may create maintenance difficulties. In the petrochemical industry, maintenance technicians are permanent, full-time employees, which affords them the opportunity to develop familarity with all aspects of an operating system. Ship owners, for the most part, do not have this luxury. Shipping firms have become highly dependent on the original manufacturers and outside field service enterprises for automation system maintenance. However, we feel that this situation can be reversed by enhancing the curriculum in the maritime academies to include additional emphasis on computer technology and system maintenance. The development of more sophisticated diagnostics for the shipboard data telemetry system can aide the operating engineer in making fault diagnosis and correction.

Production Improvements Can Cause Problems

With the advent of modular ship construction techniques in the U. S., the installation and termination of instrumentation and control wiring has surfaced as one area of production difficulty requiring further attention. Large percentages of this type of cabling require inter-module routing which dictates that complete installation cannot be accomplished until two or more adjacent modules have been assembled on the ship. Similar problems also exist in the overhaul and refurbishment of older ships. Another area of concern in today's shipbuilding industry is the ever-increasing cost associated with the installation of multiconductor, armor-jacketed cable, both from a materials and labor standpoint.

A Solution

The potential exists, through the use of time-proven SCADA technology, to monitor and control an entire machinery plant via four wires. The cost savings in cable and installation labor can amount to a few percent of total ship cost. This can easily equate to seven digit numbers.

SUPERVISORY CONTROL APPLIED TO MARINE AUTOMATION

The following section presents a phased approach to the application of SCADA technology to marine automation. As the various potential configurations are presented, an increased emphasis is placed on the capabilities of the processing system.

Phase One: Marriage Of Old And New

Figure 2 depicts one possible arrangement for a shipboard remote telemetry system.

The configuration features the following major components: a dual microprocessor-based MTU, control console, and a group of remote telemetry units at strategically selected locations throughout the machinery space. The master terminal unit's dual microprocessors with its memory, loading media, and communications

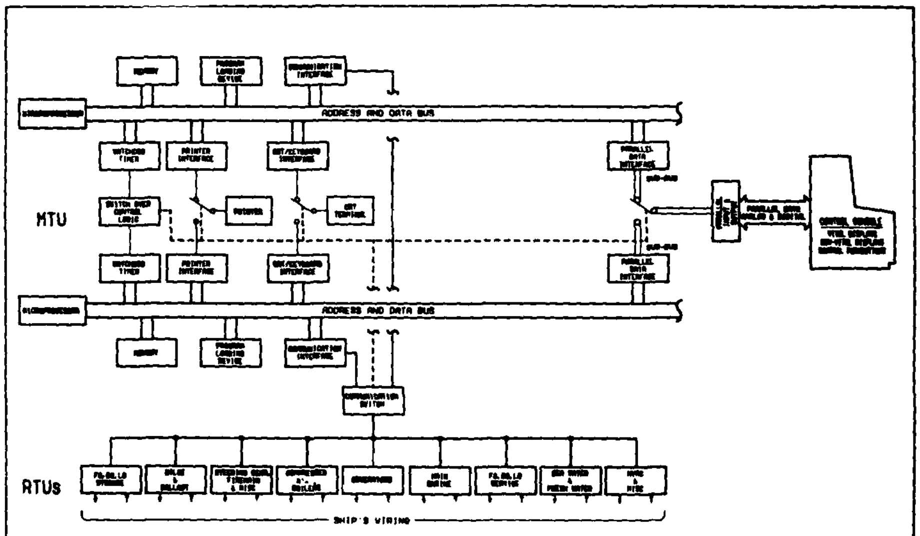

FIGURE 2. PHASE ONE

are supported by two identical buses. While each bus has its own set of interfaces, the peripheral input/output devices are common and switchable to whichever processor is in control of the system. With fully-redundant communications, one CPU can be operating a listen-only mode, thereby placing it in a hot-standby condition ready for takeover should the primary processor fail. The fail-over is controlled by a watchdog timer resident on each microprocessor's bus. This device will monitor bus activity and control switchover to the backup unit if the primary CPU should halt. This redundancy approach, typical of many pipeline systems, provides a high degree of system reliability. The controlling processor acquires data by sending out, on a pair of wires, the address of an RTU at which the desired information is located. All remotes on the same communication circuit will receive the polled address, but only the designated unit will respond with a data transmission back to the host. The subject system depicts a distribution of remote units arranged by subsystem. For example, one RTU located near the ship's diesel generators provides a focal point for the generators' instrumentation and control wiring; another RTU might be dedicated to all monitoring functions of the main engine, including bearing, exhaust, and cooling water temperatures.

The control room contains a console similar in appearance to conventional control consoles, However, this unit contains very few wires which interface directly with the machinery plant. All indicators, meters, alarm lamps, and pushbuttons are wired to a parallel data interface accessible by the controlling processor. Field data gathered by the host is output to the console via parallel lamp drivers and digital-to-analog converters. For annunciation purposes, the processor performs all of the required functions; ie, setpoint checking, time delay, hysteresis, visual flashing, audible indicators, and first-out determination. When a pushbutton is depressed for controlling action, the MTU immediately transmits a command message to the designated RTU, which in turn opens or closes a relay, thereby effecting control.

Additionally, the control system contains a printer for alarm and data logging, and a CRT/Keyboard. This device provides a central display from which the operator can examine the current state of all plant equipment and parameters. Alarm setpoints for analog processes can also be modified from this unit.

Phase Two: Less Old, More New

The system represented by Figure 3 shows an evolution away from the densely-packed console. The non-regulatory body required (non-vital) parameters have been eliminated. Only vital alarms, indicators, and meters are retained. Non-vital points are monitored and control is effected through a color CRT terminal having graphic capabilities. A few strokes at the keyboard will result in a pump starting or valve opening. The processor checks for and verifies the completion of this operation by changing the color of a graphic symbol representing this device on a piping diagram mimic.

Overall system response time is enhanced by the incorporation of additional communication lines to the remote telemetry units.

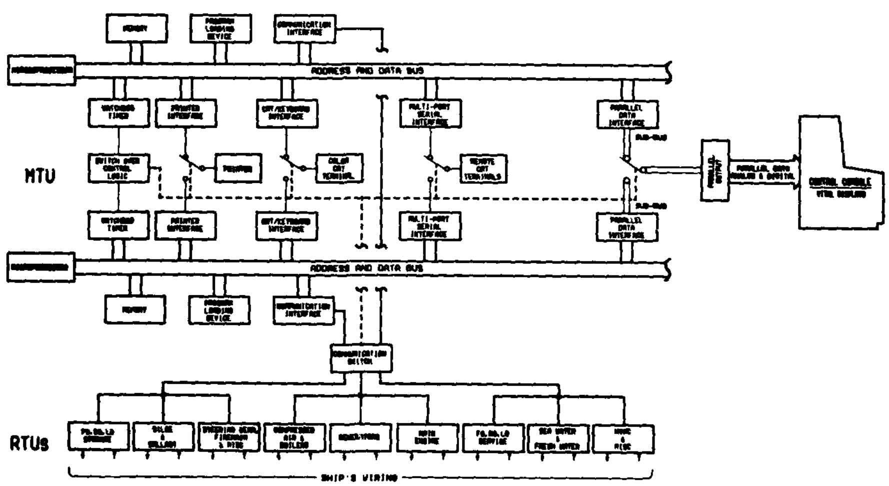

FIGURE 3. PHASE TWO

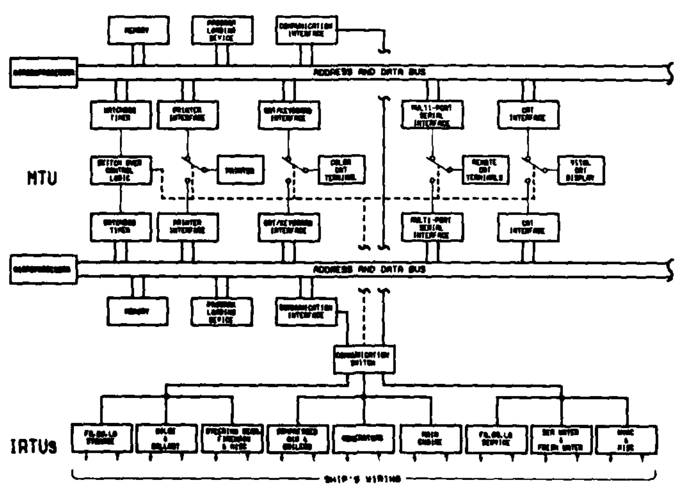

FIGURE 4. PHASE THREE

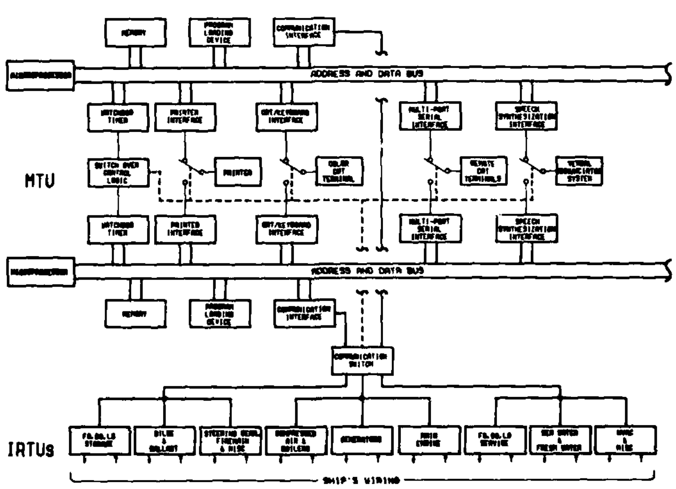

FIGURE 5. PHASE FOUR

marriage of computer and audio technology makes possible the immediate notification of any critical event. This is effected by locating speakers throughout the vessel. Advisement of corrective action to be taken is within the realm of application using processor-driven speech synthesization techniques.

CONCLUSION

This paper has presented one potential evolutionary sequence in the application of supervisory control diciplines to marine automation. Many other time-and-technology-phased schemes are possible.

Why then are we not incorporating this advanced science into ships now under construction? It is the author's opinion that the commercial marine industry tends to be very inertial. Union influences and a general reluctance to try something new have been contributing factors in stagnating commercial marine automation. However, with spiraling labor, fuel, and materials costs, I feel that the horizon is now here and the 1980's will see significant advancement in the application of computers and state-of-the-art technology to this lifeline industry.

ACKNOWLEDGEMENTS

I wish to express my appreciation to Mr. Lyle W. Ferguson, Program Manager, TANO Corporation, for his encouragement in this endeavor.

EXPERIENCE IN DEVELOPING A DIGITAL DISTRIBUTED CONTROL AND SURVEILLANCE SYSTEM.

by C.T. Marwood
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ABSTRACT

A microprocessor-based control and surveillance system for a Naval COGAG propulsion system has been designed and built, for evaluation against a ship simulation. The effect of the constraints imposed during definition and design are examined, particularly the need to minimise cost throughout the 20 year ship life. The problems resulting from applying new techniques in this field are examined, and the initial design aims reviewed. A revised system is described which uses the experience from this development to improve performance and cost effectiveness.

INTRODUCTION

The contract to develop a Demonstration control and surveillance system for future Naval ships was awarded to HSDE in 1978. Because of the significant software content, Y-ARD were appointed as independent software auditors by the Ministry of Defence (Procurement Executive), and contributed significantly to the software development and testing. A number of features new to marine propulsion were involved.

1. Specification of the system operation using Functional Block Diagrams.

2. Distributed, microprocessor based hardware.

3. Serial data links.

4. Nuclear hardness requirement.

5. Software language and structure suitable for 20-year life.

The main purpose of building a demonstration system was to evaluate these features on dry land using simulated ship machinery, so that the standard required for the first ship application could be established.

REQUIREMENTS

The requirements for this propulsion control and surveillance system were formulated over a number of years from studies carried out by the Ministry of Defence (Procurement Executive) assisted by Y-ARD, HSDE and others. (Ref 1) A typical reference ship was specified, with two controllable pitch propellers each driven by two gas turbines in a COGAG configuration. The Statement of Requirements for a distributed propulsion control system for this configuration was specified in two parts. The first part contained the overall requirements such as environmental ranges, reliability and maintainability, but also included specific directives on the choice of microprocessor and software structure.

The second part defined the operation of the system using Function Block Diagrams, without specifying how it should be implemented in terms of hardware or software. The overall system schematic is shown in Fig 1.

Initial Constraints

At the stage when the Statement of Requirements was drawn up, many of the constraints arose from the overall operation of the ship, including the reduction of current manning levels, the policy for equipment procurement, and the total 'cost of ownership' of the control system including support over the 20-year life of the ship.

These constraints and the related headings in the Statement of Requirements are shown in Fig 2. 'Availability' here is used to mean the proportion of time during which the system is operable during a mission, where operability depends on the amount of the system which is functional. Flexibility includes both ease of modification if requirements are changed, and adaptability to other control functions, or different sizes of ship. This was one of the key factors which led to the choice of the distributed system rather than the centralised controls on previous ships, which are more vulnerable to damage in a single area. The other contributing factors of increased monitoring and reduced wiring arise from the need to increase automatic monitoring in order to cut down on manpower, while cutting down ship's wiring by putting the data collection system close to the machinery. This has the consequence, however, of degrading the environment for the plant control units. Some constraints such as the selection of microprocessor and language to be used arose from the need to standardise equipment, not just on the ship but as part of an overall Ministry of Defence policy. (Ref 2)

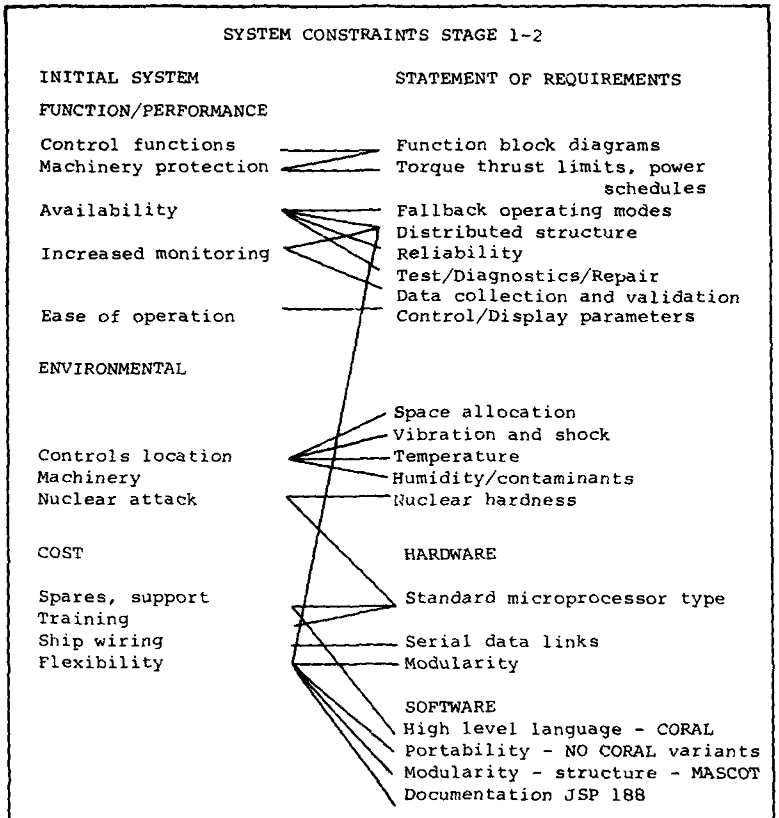

Figure 2. System Constraints Stage 1-2.

OVERALL SYSTEM DESIGN

After receiving the Statement of Requirements it was necessary to establish the hardware/software split and subdivide the software between general purpose Systems tasks, and Applications software specific to the function of one particular part of the system. This procedure was similar to the process for microprocessor-based products described in Ref 3, and consists of selecting the most appropriate way to implement the Function Block Diagrams. These diagrams largely defined the Applications software but did not identify the Systems software needed to support it.

For a flexible system providing monitoring as well as control it was necessary to simplify the hardware so that all signals for monitoring or control, could be treated in a similar manner while functions which depend on multiple signals are performed by software. The resulting split is shown in Fig 3, where the hardware function is restricted to interfacing to machinery, controls and displays, converting the signals to digital form, while all other functions such as control of scan rate, scaling, level detection and control schedules are performed by software.

The advantages of this approach are:

- Standard hardware for multiple applications.
- Minimal number of different card types.
- Minimal design for interfacing new types of signal.

The penalties of general purpose hardware are that the number of channels per card are fixed, and thus a card may not be fully utilised.

Hardware Design

The constraints arising from the Specification of Requirements are shown in Fig 4. The physical structure was determined by the environmental requirements, and a specially strengthened card frame was designed to meet the shock levels. Double Eurocard boards were found to be too resilient in spite of extra thickness, so metal stiffening bars were added. Each plant control unit is a self contained cabinet with its own processor, power supplies and control panel for local operation. (Fig 5)

The environmental constraints were also significant in the selection of components, which must not only operate at an ambient of 100°C but also withstand specified levels of radiation and electromagnetic pulse from tactical nuclear weapons. This considerably narrows the choice of semiconductor manufacturing techniques.

The microprocessor system structure was determined by the use of a standard microprocessor card, to which memory and interface cards are linked by a data bus. Space saving affected the selection of number of signals on each interface card, since a plant control unit must have a capacity for over 200 signals, which are a mixture of 11 types of input and output including frequency, voltage, current, resistance and thermocouples.

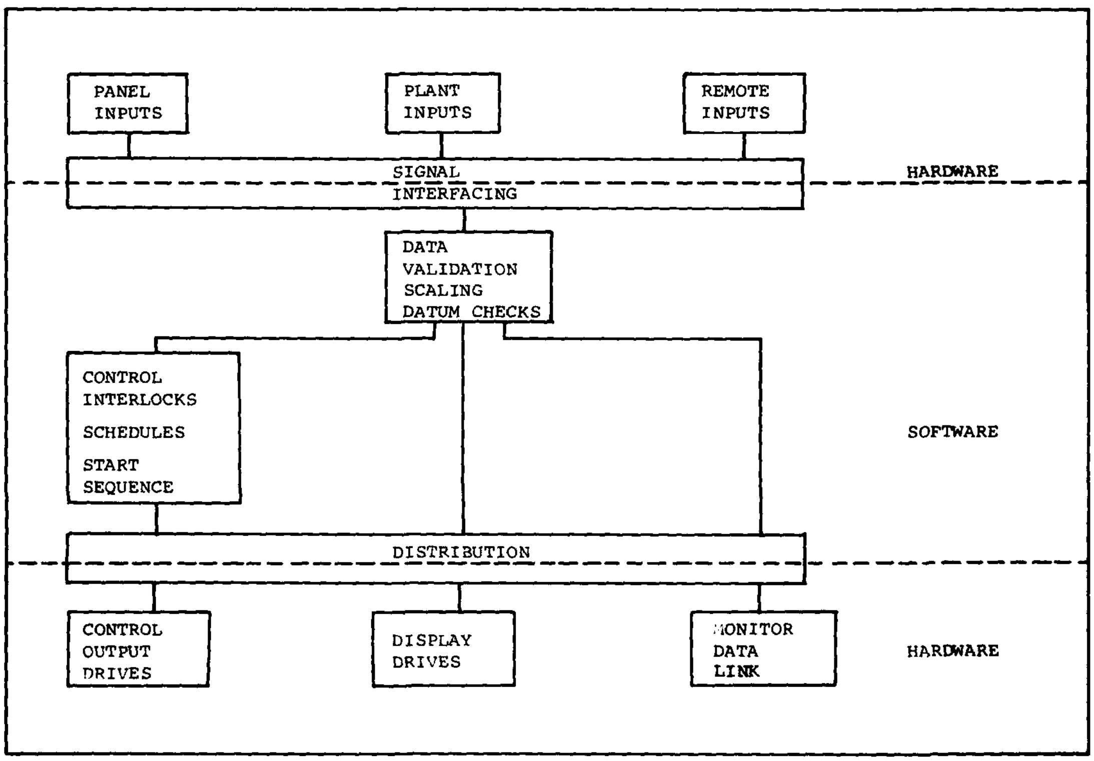

Figure 3. Hardware - Software Split.

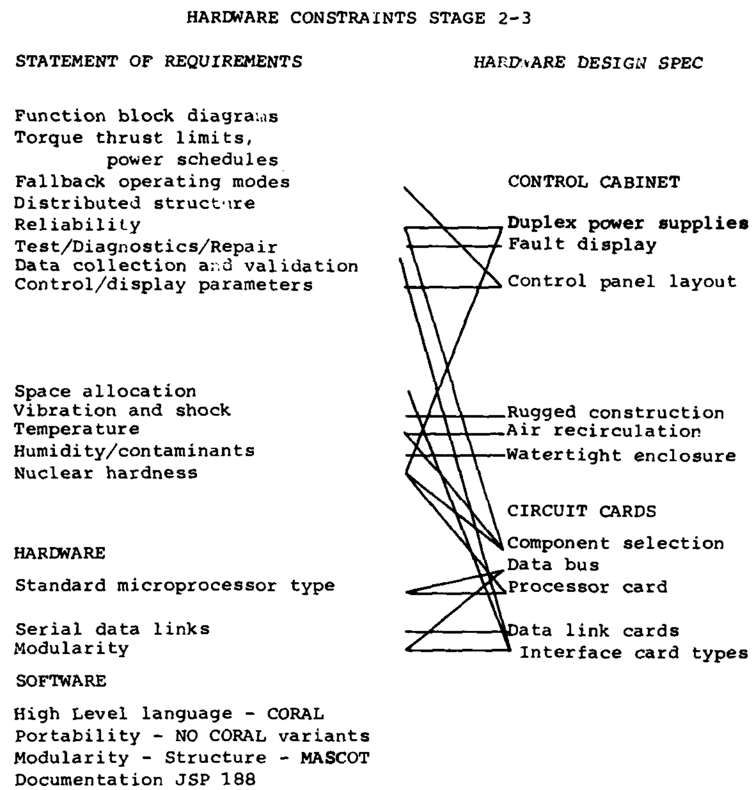

Figure 4. Hardware Constraints Stage 2-3.

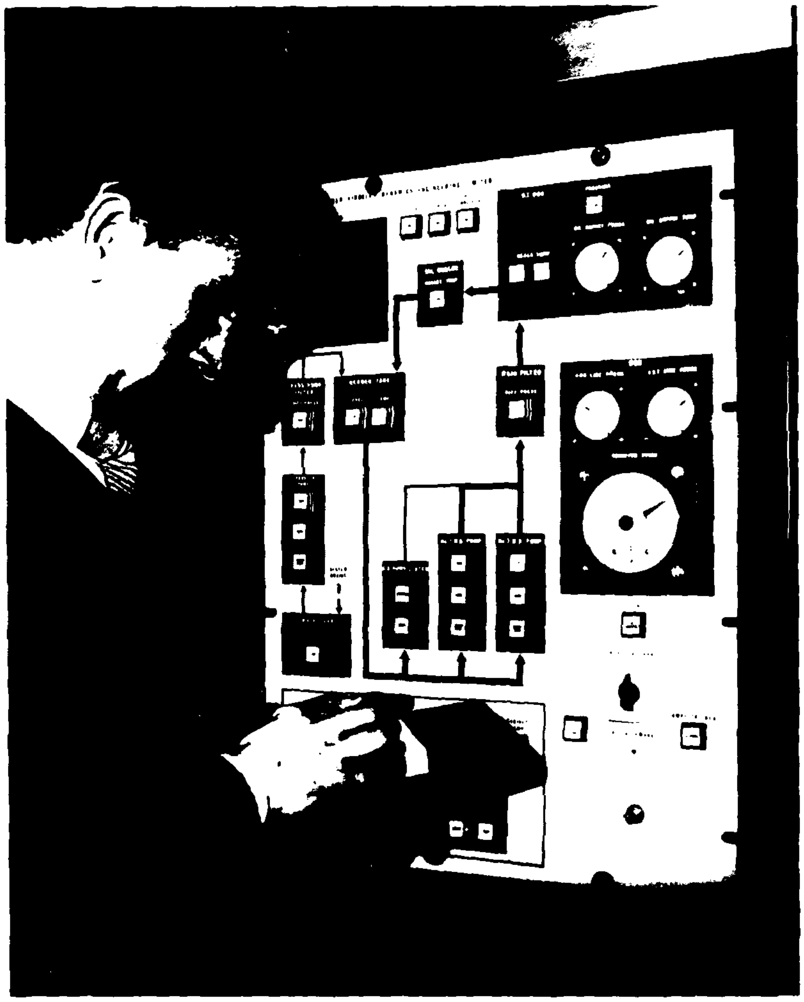

Figure 5. Local Control Panel

These are interfaced by 5 types of card which convert the signals to digital form. Special care was taken to prevent noise from reaching the data bus.

The structure of each of the seven microcomputer units in Fig 1 is similar, and varies only depending on the number of interfaces with plant machinery or other units. (Fig 6) The processor card uses the Ferranti F100-L 16-bit microprocessor and associated floating point arithmetic unit, programmable real time interrupt clock and up to 4K words of random access memory. Programmable read-only memory is used for all the program code and fixed data. On board ship this will be fusible link PROM but during development erasable PROM is fitted.

The Ferranti Serial Signalling System is used for all inter-unit links. Each unit has two or more Data Terminal cards, which convert groups of 16-bit words into message blocks by adding a header which identifies the type of message and a check code. The message is converted to serial form and transmitted down the four-wire link, which allows simultaneous two way communication. One terminal links to the system control unit and is used for sending control signals and warnings, whilst the second is mainly for surveillance data, but can also be used for diagnosis. Each data link can operate at rates of up to 3M bits/sec.

The expander enables four links to be driven from one Data terminal and is only used in the system control unit.

Further details are given in Ref 4.

Software Design

The structure and modularity of the Software largely depend on the use of MASCOT - Modular Approach to Software Construction and Test. This provides a framework of rules for modularising the software with well defined interfaces, so that changes do not propagate throughout the system. The intention is to make the control and monitor software independent of the type of processor used, and the hardware configuration. This is borne out by the constraints shown in Fig 7, where it is clear that the propulsion control software depends on the Function Block Diagrams, while only those tasks directly associated with hardware - Interfacing, Link Handler and Kernel - depend on the hardware specification.

This split between Applications and Systems software described earlier is biased towards minimising the Applications category by the overall requirements. The resulting memory requirements are shown in Fig 8.

This has been achieved by making the Data Management and Message Routing system, which arranges for the distribution and collection of information, largely independent of the number of processors in the system and the types of hardware connected.

DATA BUS
Evaluation System

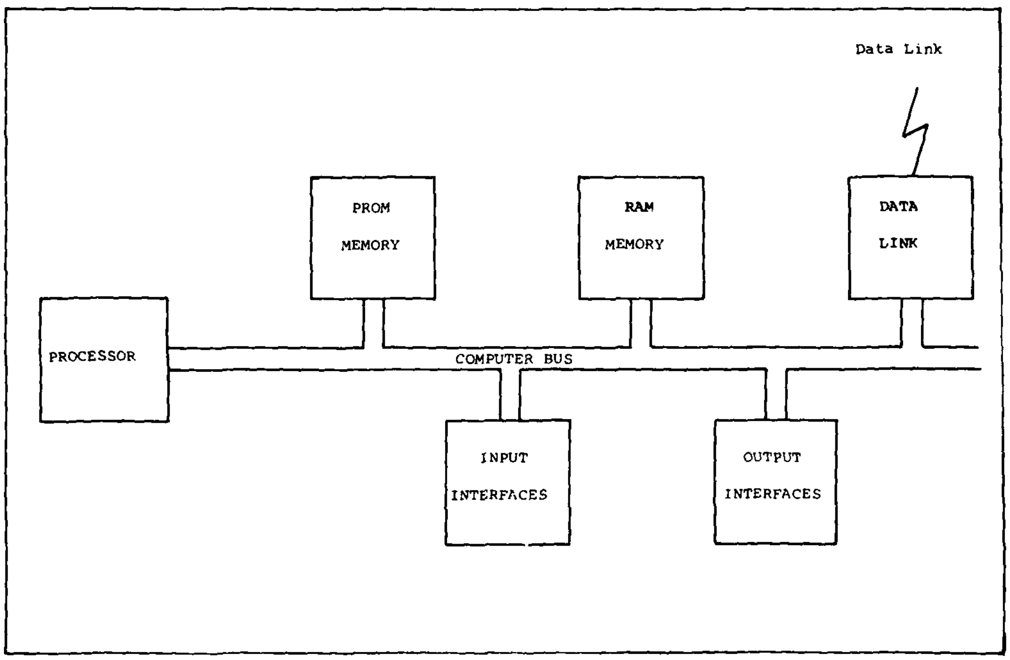

Figure 6. Bus Structure

SOFTWARE CONSTRAINTS STAGE 2-3

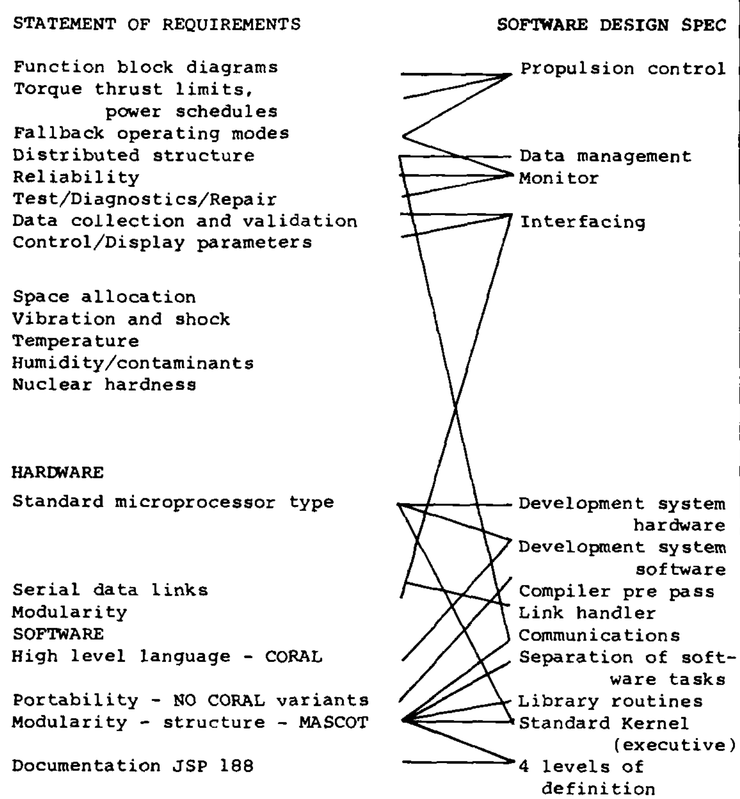

Figure 7. Software Constraints Stage 2-3.

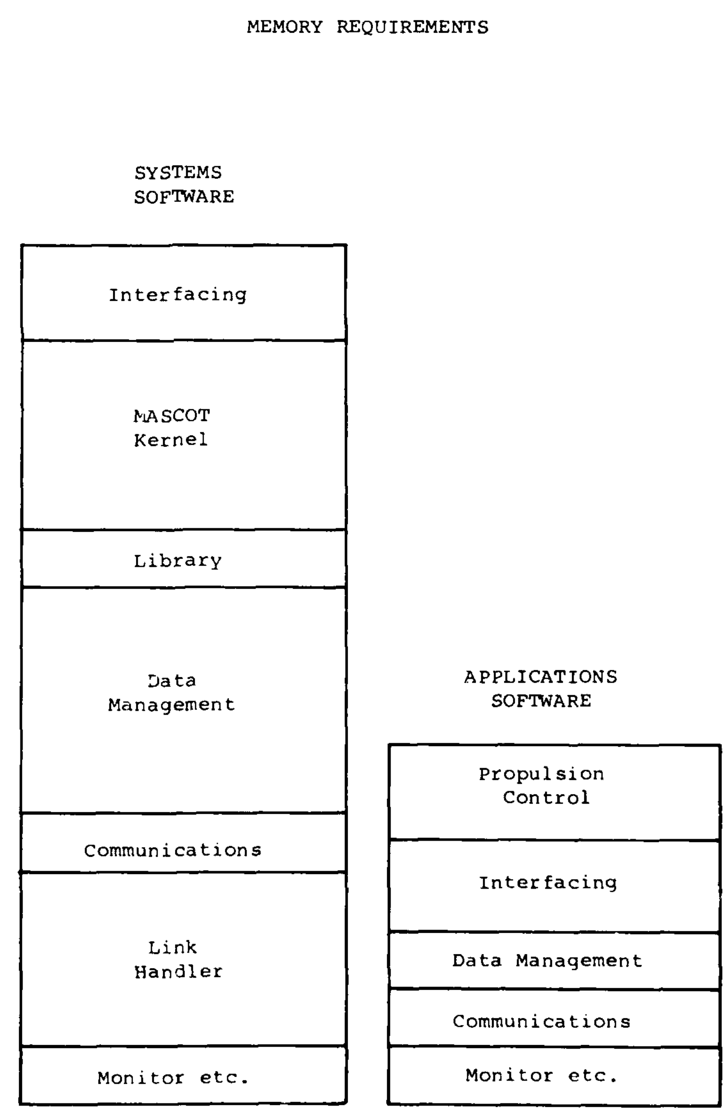

Figure 8. Memory Requirements.

Reducing the Applications software in this way minimises the cost of developing new systems but involves some increase in memory requirement and run time.

The MASCOT Kernel organises the time sharing of the processor between the software tasks by organising queues and allowing access to the highest priorities. The process relies on each task relinquishing the processor after a given time rather than forced rescheduling, and minimises queueing overheads. Software design followed the 'top down' approach of defining the top level of organisation followed by the split into tasks and then the modules of which each task is composed, before starting the actual coding in CORAL which generated the lowest of the four levels of documentation.

Testing on the other hand started from the bottom upwards, proving each module before assembling them into tasks. It was decided early on that the microprocessor development system alone would not be adequate for a project of this size where several programmers are checking, amending and compiling simultaneously, so a multi-user main frame system was set up with CORAL compiler, microprocessor simulation and other support software, linked to microprocessor development systems to test the compiled programmes. The software is loaded into PROM memory using a programming unit directly linked to the main frame computer.

INTEGRATION AND TEST

The policy of 'bottom up' testing was followed in the hardware as well as software, using the Ferranti Development System (FDS) for proving individual cards before assembling a complete processor system. The FDS consists of a card frame with resident processor, memory and interface cards linking to VDU and disc handler, with spare slots for inserting the card under test.

During the next stage each of the Plant Control Units (PCU) was fitted with tested cards, and down loaded with test software from an FDS using a serial data link. Special test equipment was built to provide input and output signals equivalent to the plant machinery, so that each of the interface channels could be exercised and the overall function of the PCU checked. As no in-circuit emulation was available, the only means of direct communication with the processor was via a handset keyboard display using machine code. Once the test software was loaded and running a VDU or FDS could be linked to the system and used for memory access or software interaction.

In parallel with PCU testing, the communications link software was tested separately, using FDS units with serial data link interface hardware. Once this was proven the testing of the System Control Unit (SCU) hardware, which is the star point of the communication system, could begin.

Having proven the hardware and that part of the software which controls the function of each unit as a free-standing processor, the major task of testing the overall communication and co-ordination of the system began.

Due to programme constraints of time and effort it was decided to proceed with connecting some of the PCU's to the plant simulation at the evaluation centre while the overall software proving continues.

At the time of writing, three PCU's have been linked to the simulation and prelimary tests have been successfully completed.

DEVELOPMENT PROBLEMS

The principal problems arose from the impact of the irresistible torce of expanding software meeting the immovable object represented by the 32K address limit of the processor. The immediate effect was diversion of effort into optimising memory requirements and delays in testing because there was insufficient space for diagnostic routines. A paging system was devised but has not been adopted because of the associated changes needed to software development facilities and program structure.

The increase in software overheads has also affected response times both within the PCU cabinets and in the communications system which links them. The communications hardware deals with the lowest of the four levels of control by carrying out the message framing and cyclic redundancy checking, but the remaining three levels including master/slave relationship and retransmission are controlled by the microprocessor. This takes up a significant share of processing and memory resources, particularly in the SCU which handles multiple links. Further development on software optimisation and tuning is expected to give significant improvements.

LESSONS LEARNED

Functional block diagrams were found to be useful for defining the overall system operation, but when expanded to the lowest level with hundreds of input and output signals they were difficult to interpret because of the differences between the organisation of the data in the diagrams and the hardware layout. It should be appreciated that the diagrams only represent the control functions which are a small proportion of the overall software. They are also useful only as long as they are kept up to date, which is a task best performed on a word processor.

Much of the response delays arise from using general-purpose software routines, designed for total processor and configuration independence, to carry out simple tasks which need rapid execution such as I/O data scannning and communications protocol. While this is a desirable long-term aim, it is difficult to achieve given the memory and speed constraints of present day hardware. Re-examination of the overall data collection system shows that many of these tasks can be carried out much faster and cheaper using hardware, without losing the flexibility to add channels or modify scaling values.

Designing the hardware to withstand the specified levels of nuclear activity has had more extensive effects than were originally foreseen, and the increase in development and build cost is significantly more than the difference in component prices. There were very few suitable microprocessors available, of which the F100L was by far the most suitable.

However, support chips, memory and other devices have had to be chosen from a restricted range of bi-polar and other hard technologies rather than third generation VLSI, resulting in a comparatively low packing density and high dissipation. Special power supply regulators also had to be designed rather than using low-cost general purpose units.

The bus linking the processor, memory and I/O cards was chosen to be the F100L processor bus, and space was allocated on each card with the intention of standardising on the MODBUS system in line with Ministry of Defence computer policy. The penalties of this approach are that a significant area is dedicated to interfacing, using high-dissipation devices whose wide range of operating modes is under-utilised.

In spite of these problems, a viable system has been built and much useful information gained by trying out new approaches in hardware and software and measuring the results.

All software, except for a few lines of code, has been written in a high-level language, (CORAL 66), and can easily be transferred onto other types of processor. The number of types of card has been minimised, reducing build and spares costs, and a highly reliable supply voltage regulator with proven immunity from RFI and EMP interference has been developed.

REVIEW OF REQUIREMENTS

Before starting the development of the revised system, HSDE made a careful review of the requirements to assess needs of Naval Ships throughout the world.

The initial design aims at the time of designing the Demonstrator system can be summarised as:

Flexibility

Cost Effectiveness

Long life software

Ruggedness

Maintainability

Performance

These are all still necessary, but as in any multi-variable problem the solution depends on the emphasis given to each requirement.

Flexibility.

The ability to change configurations has been emphasised to the point where it was felt desirable to be able to plug additional plant units into the system without changing any software. It should certainly be possible to add or amend individual control and monitor signals by onboard changes, but an extra processor would generally imply machinery changes requiring a refit, during which PROM memory cards could be replaced by modified versions containing additional software, with minimal cost and effort.

Cost Effectiveness

Since this system replaces separate control and monitoring systems used at present, significant savings in equipment and cabling costs have been made possible. However, the initial cost on smaller ships needs to be brought down to compete with analogue controls.

Long Life Software

Processor independent software is essential for the upper level systems functions, particularly where changes are likely, to ensure that hardware obsolescence does not require a major software rewrite. However, the hardware/software split should be made such that simple routines which affect performance or response time are performed quickly, either by hardware or low-level firmware. The data I/O system and inter-processor communications link should also be independent of the type of processor used.

Ruggedness

The techniques suitable for machinery space electronics are well known to designers of engine-mounted systems, but the combination of high temperature and nuclear hardness greatly restricts component selection with corresponding cost increase. Minor changes to specification can significantly reduce space requirements and heat dissipation, as well as build and development costs.

Maintainability

A microprocessor system should be serviced as a set of black boxes, with a simple diagnostic system to identify the box to be exchanged.

The diagnostic equipment should be available with the first hardware, and should also check the interfaces between the black boxes.

Performance

The trend towards reduced manning increases the number of points to be monitored, and the need for dynamic data recording demands a fast scan rate. Future ship systems must offer a high performance data collection capability.

THE WAY AHEAD

Revised Plant Control Unit

In deciding the best way to implement the revised requirements, it has been important to use as much as possible of the hardware and software experience from the Demonstrator system. HSDE have decided to go ahead with the development of a revised system which overcomes the problems identified earlier, and also offers a number of other advantages. This system which is described more fully in Ref 5, is equally suited to monitoring, control or a combination of both. The principal features of this system are:

- A simplified input/output bus, completely separate from the processor bus, to which the I/O cards are connected. This reduces the amount of system electronics on each I/O card.

- A scan controller which uses the I/O bus to collect data, convert it to digital form and store it in memory for processor access.

- Separate control and communications processors, handling the data conditioning and validation, and the serial data link protocol respectively.

Fig 9 shows the structure of the system, with the scan controller forming the link between the processor bus and the I/O bus. This makes it possible to change to an alternative processor without altering the input/output interfacing cards. In a market where the choice of processor may be determined by customer policy, and obsolescence is likely to be a recurring problem during the lifetime of a ship, this facility is very desirable.

The I/O bus itself has been designed for high scanning rates, but can also be operated slowly by switches for diagnosis. Because scanning is sequential, the input and output data rates are simple to calculate.

The control and communications processors are designed as a two card module, which includes the PROM and RAM memory for the unit, and thus replaces four cards in the previous system.

Revised System Control

The System Control Unit which co-ordinates the Plant Control Units uses similar hardware to the Master Unit in Ref 5, consisting of a double communications module capable of driving up to 20 serial data links, and a control module with its own processor and memory which is dedicated to the control and monitoring function. The increased processing power overcomes the problem of communications delays caused by peak data traffic on the monitoring system.

In a stand-alone propulsion control and monitoring system the SCU is linked to control panel units at the Ship Control Centre and Bridge as in Fig 1. However, since the communications system capacity is greatly increased, the SCU collects all the monitoring data as well, and provides a single output to the monitor and display system, replacing the separate serial links from each Plant Control Unit previously required.

Data Highway Connection

If a ship Data Highway is available, the propulsion control system should be attached to it via the SCU as shown in Fig 10, rather than the conventional manner of linking individual units directly to the highway. There are several arguments for this configuration:

1) A fallback control position can be provided at the SCU as shown. With a conventional highway this would require an additional processor and highway node connection.

2) Only one highway node is needed to attach all the Plant Control units. As there are several types of highway, this system can be attached with the minimum of hardware and software interfacing, since the control commands at the SCU input are very simple.

D A T A B U S
Modified System

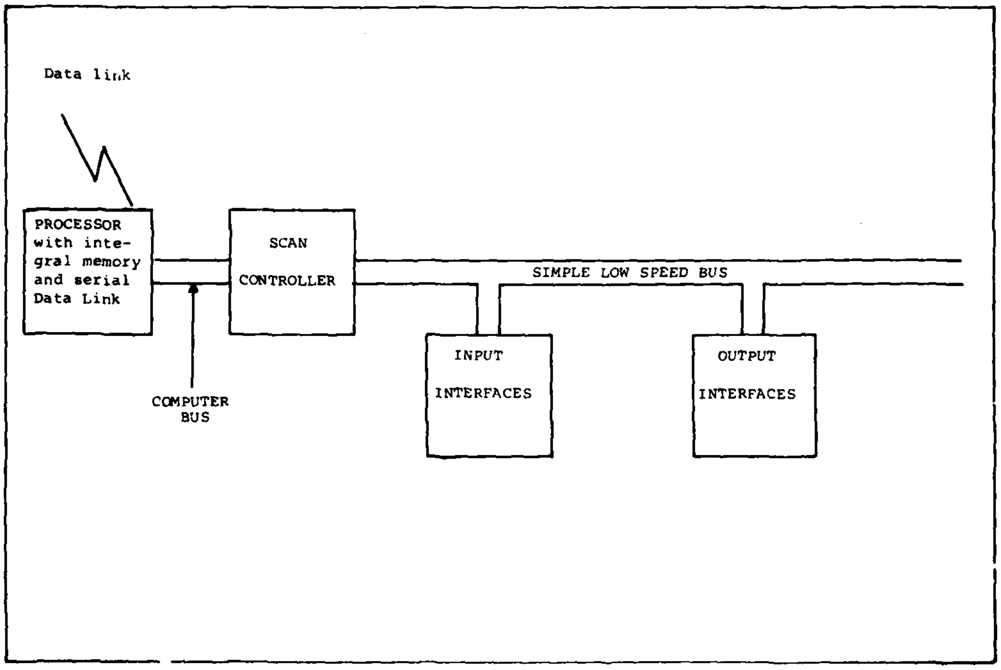

Figure 9. Modified Bus Structure

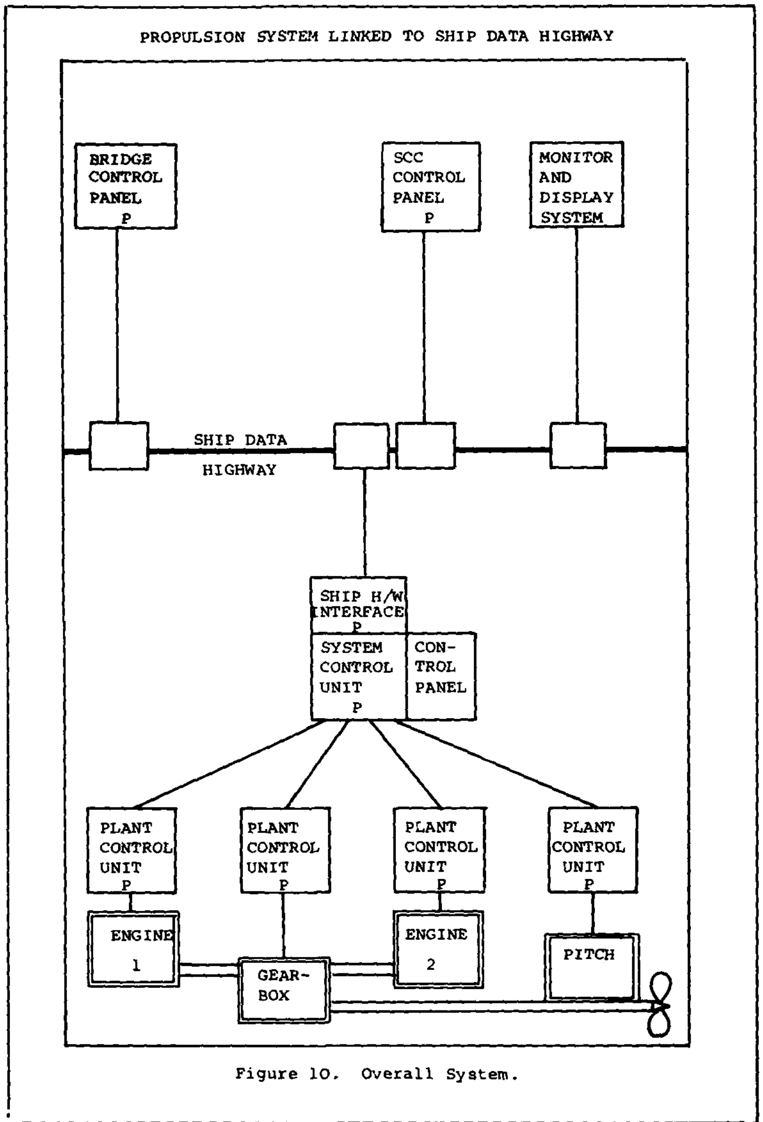

Figure 10. Overall System.

3) A standby SCU can be added for maximum integrity, or fallback increase/decrease control direct from the SCC to each Plant Control Unit can be provided using a new link designed to fail set for open circuits or any cross connection of the three control wires.

CONCLUSION

The development of a distributed, microprocessor-based system combining control and monitoring, brought together for the first time a number of techniques new to the marine field, with an additional requirement for nuclear hardness. Ineveitably there have been new lessons to learn, many of which would never have emerged from paper studies. An examination of the constraints during the development shows that where multiple constraints converge on one design activity, the risk factor and development cost are likely to increase. A review of requirements and development problems has led to a revised system which overcomes the major difficulties and employs more cost-effective semiconductor technology.

The Demonstration project has already achieved much of its purpose, which was to evaluate the new features of the system and thus reduce risks on the next ship programme. These features can therefore be used with confidence on future ships. The revised system is offered by HSDE as an alternative for less exacting nuclear environments, making use of the experience gained and providing an improved data collection capability.

ACKNOWLEDGEMENTS

Acknowledgements are due to my colleagues at HSDE for their dedication to the project and for advice and assistance on this paper. Opinions expressed are those of the author.

REFERENCES

(1) I.W. Pirie and J.B. McHale, "Evaluation of Digital Technology for use in Naval propulsion control systems", Fifth Ship Control Systems Symposium, Annapolis, USA, 1978.

(2) R. Whalley, "The formulation of a computer strategy for Real-time, Shipborne, digital systems", Fifth Ship Control Systems Symposium, Annapolis, USA, 1978.

(3) D. Aspinall, "The microprocessor as a component and tool" I.E.E. Electronics and Power Journal Nov.1979.

(4) C.T. Marwood, "Experience in development of a Naval distributed control system", IEE Colloquium March 1980.

(5) M.J. Curran "Monitoring Systems", Sixth Ship Control Systems Symposium, Ottawa, 1981.

THE CONTROL OF NAVAL CONTROLLABLE PITCH PROPELLERS

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INTRODUCTION

It is the purpose of this paper to indicate what can be achieved in respect of pitch control and to give some guidance on the degree of complexity which results.

The CPP's of destroyers and frigates commonly cost a great deal more than those of large car ferries. Whilst writers of naval CPP specifications sometimes state that complexity of design should be avoided, their specifications quite often make this desirable end difficult to achieve.

There is little between the requirements for a warship and those for a large car ferry - indeed the latter will be more demanding in respect of precision of pitch holding when going astern. Until fairly recently, the warship required more elaborate blade design, but now the ferry also requires elaborate blade design for somewhat different reasons - the comfort of passengers and crew accommodated right aft. So at first sight, current standard commercial CPP's will suffice for frigates and destroyers, and indeed some of the smaller of these ships are so equipped.

There are however penalties, essentially in respect of weight rather than accuracy, if standard commercial CPP's are fitted to warships. This is largely due to the different amount of shaft boring considered appropriate.

PITCH ACCURACY

It is necessary, both to achieve the desired power absorption and for military reasons, that naval CPP's hold pitch repeatability within $\frac{1}{2}^{\circ}$ or so, and desirable that a better figure - say $\frac{1}{4}^{\circ}$ - be obtained in all circumstances when cruising at sea. In contrast, the accuracy of pitch when the ship is manoeuvring in harbour is significant only around the zero thrust position. At other angles $\pm 1^{\circ}$ might well suffice, though there is no difficulty in doing better providing the rate of change of pitch demanded by the control system is within the output of the CPP hydraulic pumps and system.

Table 1 compares single acting and double acting propellers in respect of pitch changing ability, the four types of propeller chosen being those manufactured by the author's company. The reduction in

Table 1. Comparison of Pitch Changing ability, Single Acting and Double Acting Propellers.

Hub Dia. mm	Model	Oil Pressure	Nominal turning Moment on Crankring KNM	Approx. nominal % loss due to friction at Zero thrust	Effective Moment on blade KNM	Bolt array bending modulus cm^3
870	XX 4	375 psi	49.5 Max.	35	32	1750 approx.
		650 psi	85.8 Max.	35	55.8	
880	XL 4	1800 psi	57 Min.	69	17.7	1520
860	XK 4	500 psi	41 Max.	61	16	1480
860	XS 4	600 psi	29 Max.	57	12.5	1226 (blade journal)

Note: 375 psi is approximately the pressure utilised in Type 42.

650 psi)
1800 psi) is approximately the pressure the standard design is considered for, though the
500 psi) XX and the XS are sometimes over-rated for specific applications after suitable
600 psi) attention to detail.

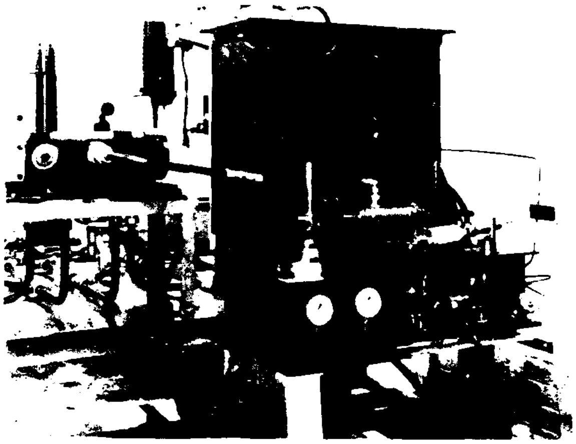
frictional losses in the hubs of double acting propellers are startlingly large - sufficient to be of importance when selecting which type of propeller to fit. In fact, only double acting propellers could meet the manoeuvring specification demanded by some navies in the last decade, i.e. ability to meet any demanded pitch angle at any ship speed and at up to full propeller rpm. The first ships - to the author's knowledge - in which this severe requirement applied, were the Vosper MK5 and MK7 frigates for Iran and Libya, ships with over 20,000 SHP per shaft and propellers rotating at more than 400 rpm at full speed. Now that more moderate propeller conditions are usually selected, single acting propellers will meet the actual manoeuvring requirements of most warships, saving money in first cost, despite their requiring more pumping power to overcome frictional losses. The additional friction of single acting designs does, however, impose a delay in response to controls demands, and the significances of this delay requires assessment if dynamic response is to be given serious consideration.

A fairly typical figure for the pressure change required to overcome frictional effects in a large 500 psi single acting CPP on the test bed is 65-70 psi, and for a 500 psi double acting propeller is only 30 to 38 psi. It will be seen that whatever control system is adopted the delay time in establishing a change of motion in the single acting propeller might be approximately twice as great. This disadvantage will also apply to single acting propellers of the trunnion bearing design, though not to so great an extent.

Taking a typical frigate with double acting propeller, it is likely that the pressure change required will have to be effective throughout a volume of some 70 Imperial gallons of oil if inboard control is used and 30 gallons if a hub control valve is used. In the latter case the entire 150' long valve rod has to be set in motion by an auxiliary servomotor subject to its own time delays, so that the time delay between an order being received at the inboard main control valve in the one case or the auxiliary servomotor control valves in the other and propeller blade motion is likely to be between .5 and 1 seconds in either case. With a single acting propeller the overall delay - at full rpm on a test bed installation - is about twice as great.

Performance at sea is, however, of more interest than test bed performance, and performance at sea is assisted by the fluctuations in load induced by the ships wake. Friction coefficients of propeller bearing material pairs average around .17 (Figures 1 and 2 show measurement machines for friction and wear rate), but even under heavy loads power losses calculated using this figure are not met with in practice since the propeller is always subject to quite large cyclic fluctuations in water flow and these reduce apparent friction. It is indeed practically impossible to propound a reliable average figure. The better designed the ships hull and A frames, the less fluctuation in loading occurs to each blade in passing through the various wake shadows which it meets in the course of each revolution.

However, a figure of .12 instead of .17 is sufficiently conservative for use in calculations for fine lined twin screw ships. This leads to the delay times found at sea being substantially less than those met with on the test bed (with average sea loadings simulated by centrifugal force on loading arms). Figure 3 shows a double acting propeller on its test bed. Figure 4 shows a similar test bed for single acting propellers.



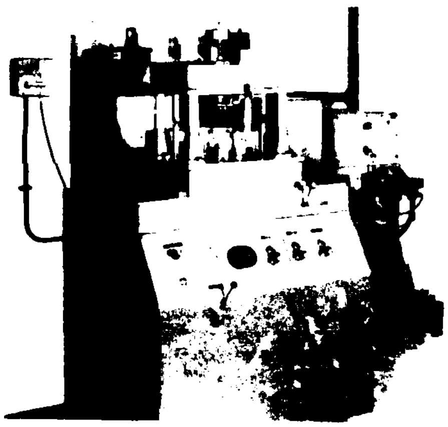


Figure 2. Friction and Wear Rig.

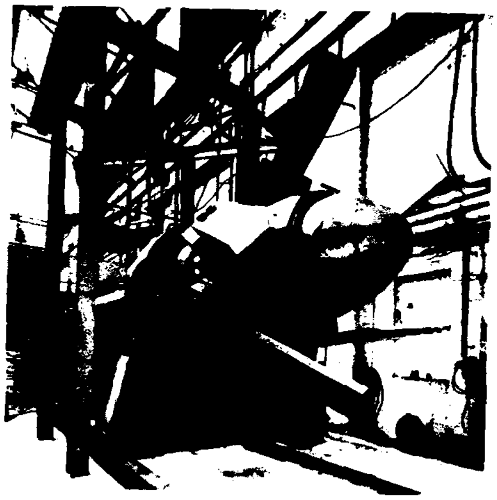


Figure 4. Commercial Propeller Test Rig.

It may well be asked, however, whether a delay in response of as much as 1 or 2 seconds would be significant in manoeuvring a ship displacing around 4000 tons and able (say) to provide not more than an average stopping force of 150 tons per shaft during a manoeuvre from 30 knots to zero.

In practice, the emergency stopping time of the ship (from full speed, full power) of nearly 30 seconds is unaffected by the addition of a delay of between 1 and 2 seconds in initiating propeller movement - since a delay of this order is needed initially to reduce the input of fuel to main gas turbines sufficiently to make it practicable to change pitch without risking overspeeding and tripping the engines.

Similarly, the opposite manoeuvre of accelerating a frigate from stop to full speed is in practice entirely dependent on the rate of acceleration which it is sensible to apply to the ships gas turbines. Both single acting and double acting CPP's will commence moving ahead from their zero thrust position well before the rate of engine acceleration can become significant. The rate of pitch movement thereafter towards design pitch can readily be programmed to suit the gas turbine. The CPP pumps, within certain limitations which are discussed later, can be sized accordingly. So whilst it might be thought that the CPP design would be critically affected by delays in response and by control rate requirements during emergency stops and emergency acceleration - in practice it is not so. The stresses set up in propeller and shafting most certainly are important, but control requirements are not, always providing that servomotor power is adequate to change pitch comfortably within the pressure limitations of the system fitted. It is, of course, possible to devise optimum programs for pitch/throttle positions throughout both the emergency conditions postulated, and a degree of advantage results. It is nevertheless also possible to provide in a control room merely one lever to control throttle position and another to control propeller pitch. An engineroom artificer or mechanician with E.R. ticket should prove capable of manoeuvring his ship without any controls interconnecting the two, given training and adequate (but simple) instrumentation. It is indeed very desirable that good hand controls should be available for throttle and pitch, for in action electric leads and control cabinets are far more susceptible to splinter damage and to fire than were the control rods used some years ago.

LIMITATIONS ON PITCH ACCURACY

We have seen that pitch control during manoeuvring does not necessitate very great precision. Pitch control in action and indeed whilst cruising is a different matter. It is important to know what pitch is actually set on the blades and to be able to alter this exactly as required for a number of reasons, including efficiency.

Naval CPP's now on offer have their main servomotors in the propeller hub, either directly beneath the blade palms in the case of high pressure single acting designs (see Figure 5) or further aft in medium pressure single acting designs. Double acting hubs vary a good deal in internal arrangements (see Figure 6 for Stone Vickers' XX design) but all major designers of both single acting and double acting CPP's use - so far as the author is aware - either one tube or two concentric tubes to the moving parts of their main servomotors and use the fore and aft motion of these tubes, as pitch changes, to indicate the extent of that change at the oil transfer box. It therefore follows that direct measurement of the relative position of the forward end of the tubes against some fixed point provides the datum for both local and remote control by whatever means - electrical, mechanical, hydraulic or pneumatic - are preferred.

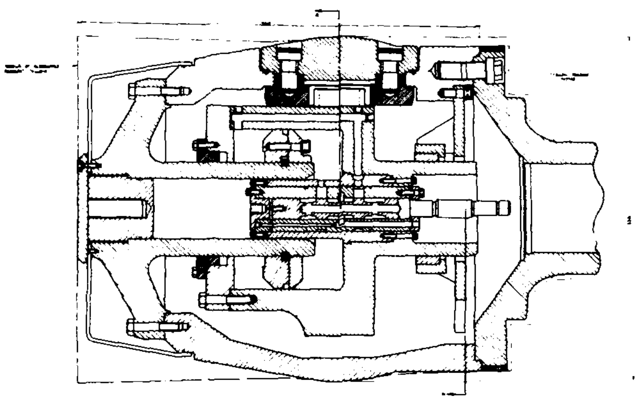


Figure 5. XL5/125 Propeller Hub.

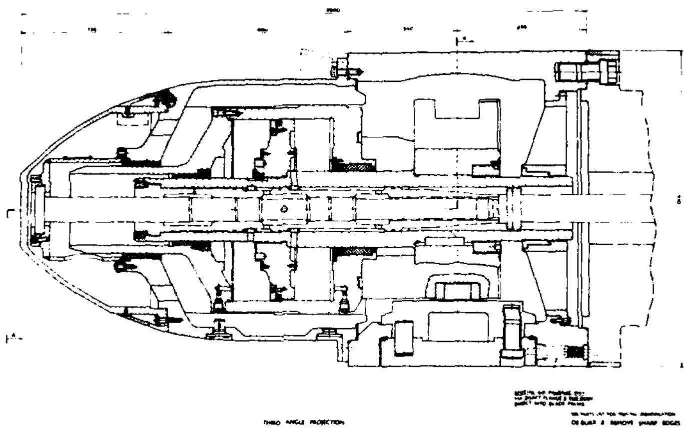


Figure 6. XX 155/5 Propeller Hub.

At this point a digression is necessary to explain why direct electrical measurement of blade pitch is not resorted to. It will be appreciated that failure of pitch indication at the propeller entails docking the ship. There is therefore a predisposition for the engineer to rely on the most massive moving parts of his mechanism to transmit blade position. Those parts are necessary to the functioning of the mechanism, and it is merely a matter of logic to use them additionally to provide blade angle data.

There would, however, still be a case for using electrical means for transmitting information if there was any prospect of such information being more accurate than that taken from the forward end of the O.T. tubing; providing that the life of the potentiometer, its drive and its connections working immersed in oil under pressure and under conditions of heavy but, in part, unpredictable vibration were adequate, and that accuracy could be maintained over a period of years in transmitting the signals which resulted from the rotating shafting to the ships structure. There is at present no certainty of meeting any one of these conditions, not at a reasonable price.

In contrast, mechanical transmission is known to be reliable and it is possible to correct most of the inherent errors in mechanical transmission of blade position by attention to detail in design.

The double oil tube arrangement necessary for propellers with inboard control is used by Stone Vickers to provide differential feedback and hence improve pitch accuracy. Most CPP designers compare the transmission shafting forward end with the lateral position of the oil tube free end to determine pitch, either ignoring or compensating for the fact that the propeller shaft length varies with thrust due to shaft compression and moves across the thrust block clearance at zero thrust (inducing a $\frac{1}{2}^{o}$ pitch error). In single screw ships with short shafting shaft compression can be insignificant, but it is a different matter with a twin screw destroyer, with shafts around 150 feet in length. Typically, such a ship may have a servomotor total stroke of 12" and a shaft compression of 2 to 3% of this at nominal full speed thrust when both shafts are in use. There are circumstances when the normal thrust/ships speed relationship does not hold; when accelerating, when power is varied between shafts or one shaft is trailed. This makes compensation difficult. Now the oil tubes are not subject to thrust, and if carefully designed, the effects of Poissons ratio on their length cancel out the length increase due to pressure. They are therefore the ideal means of transmitting hub servomotor movements to the free end of the shafting where electronic pick offs are situated. It is believed most unlikely that more accurate results could be achieved by siting the electronic sensors in the hub, even if the expense of doing so were ignored.

The maximum variation of achieved Propeller Pitch in a Frigate CPP System

The overall accuracy of propeller pitch control is related to the cumulative errors in the actuator/controller, control valve and feedback linkage of the system. It can be measured in terms of repeatability, defined as follows:

Repeatability - The ability consistently to return, within a given tolerance to a given achieved pitch for the same set of external conditions; these to include pitch demand and shaft speed.

Tests have been conducted on the Stone Vickers full scale shore test facility which represents, in most particulars, the propeller system of a Type 42 destroyer fitted with open circuit hydraulics. This demonstrated that a repeatability of $\pm$ 0.3° of blade pitch can be achieved under all normal operating conditions. It should be noted that the propeller, shafting and O.T. box design is of 1970 vintage. The class is still in production, and for reasons of interchangeability the equipment has not been altered significantly since that time.

By fitting the propeller system with additional transducers, measurements were made to identify the magnitude of the errors in different parts of the system.

(a) Actuator/controller deadband + gearing backlash = 0.08°. (Defined as the change in pitch demand to cause the valve spool to reverse its direction of movement.)

(b) Control Valve deadband = 0.18°.
Defined as the reverse movement of the spool which causes a pressure difference sufficient to maintain the achieved pitch but not to change it.

(c) Hub backlash = 0.04°.
Defined as the change in pitch demand, during a small reversal of blade pitch, between the times when the control valve spool reaches its final position and the feedback system shows the hub to have started moving.

(d) Repeatability = $0.08 + 0.18 + 0.04 = 0.30^{\circ}$.
Defined as the change in pitch demand, from the start of a pitch reversal, to the start of a change in pitch achieved: Repeatability = sum of (a), (b) and (c) above.

For any future new design, it is considered that these sources of error could be further reduced, if necessary, by attention to particular details of the system.

In future:-

(a) The position transducer providing feedback within the Actuator/Controller should have a linear response.

(b) Control Valve errors can be reduced by optimising the gain of the spool assembly.

(c) The pitch feedback system should fully compensate for non-linearities in the hub mechanism.

(d) Hub backlash can be minimised by deleting bearing assemblies requiring working clearance from the pitch feedback mechanism. This entails changing the pitch locking mechanism.

Use of a hydraulic system for propeller control dictates that there will always be some pitch following error in the transient state. The magnitude of such an error is a function of control valve spool design combined with the pump output. Typically, the error is a maximum of 2° between the demand position of the actuator output shaft and the hub; the largest errors occurring at maximum propeller loadings when large control valve spool movements are required to achieve the required pressure differential at the hub.

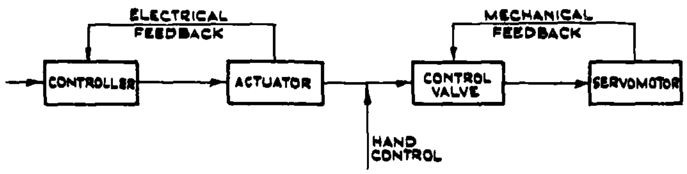
It will be realised that to pass the sizeable volumes of oil involved in pitch changing, pump pressure must rise much more than does the propeller pressure whether an inboard valve is used or the control valve is in the hub. Pressure loss in a typical destroyer system changing pitch at over 3°/sec. can be 20 atmospheres. Porting restrictions account for part of this pressure loss when open centre control valves are used. When variable delivery pumps are fitted, the loss is largely a matter of frictional effects and compression as two long columns of stationary cold oil are set in motion. The loss is, of course, taken into account in assessing pump power.

Feedback Arrangements

In a recent investigation for a potential major warship installation, Stone Vickers investigated a number of alternative feedback arrangements. The first of those outlined below was the current company standard system.

Four alternative systems have been considered:-

(a) Actuator Position Feedback



The control system loop is closed by a feedback signal from the actuator position.

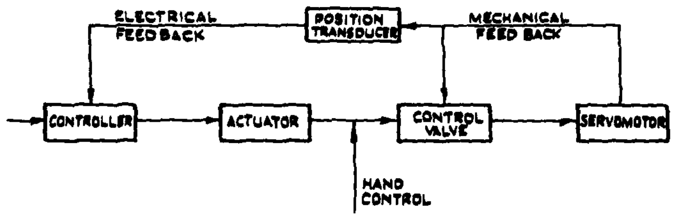
Disadvantage. Errors in both controller and servomotor closed loops are cumulative, i.e. pitch error resulting from control valve offset when holding pitch is not compensated for by the control system. Pitch repeatability as defined above is not affected. This offset is however minimal when the propeller is at design pitch and ship speed is appropriate to the shaft rpm in use.

Advantages. The number of interfaces between the control system and the hydraulic pitch setting system is minimised simplifying installation and fault diagnosis.

A simple and accurate local hand operating facility is readily incorporated.

The philosophy is consistent with T.42 Open Circuit Hydraulic System which has been shown to meet the accuracy requirements under nearly all conditions on the test bed. Its performance at sea is discussed later.

(b) Servomotor position Feedback from mechanical Feedback Linkage.



Disadvantages. Increased interfaces between control system and hydraulic pitch setting mechanism.

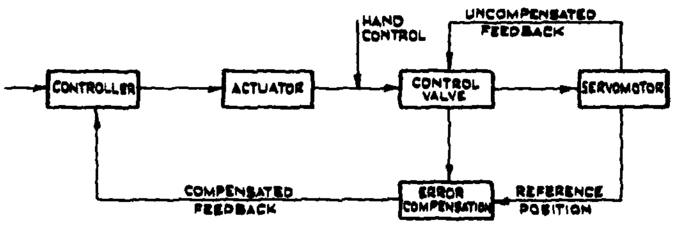
No reduction in mechanical complexity possible over (a).

Possibility of instability.

Advantages. Accuracy of pitch setting improved, i.e. spool valve offset compensated for in holding pitch condition and effect of backlash in actuator drive reduced.

Local hand control available as in (a)

(c) Servomotor position Feedback using simple mechanical Feedback correcting for Shaft Strain and Temperature change electrically.



Disadvantages. Mechanical pitch indication will not be compensated for feedback error and accuracy of local emergency hand control is degraded under load, though still accurate as a datum in the static condition.

Increased interfaces between control system and hydraulic pitch setting servo.

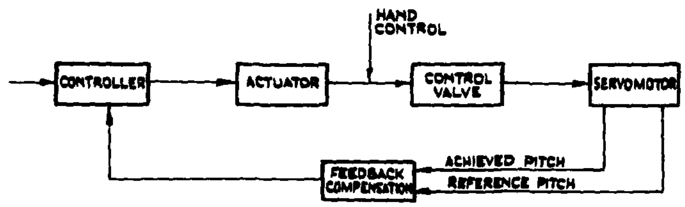
Relies on transducers buried in O.T. box, reducing maintainability.

Advantages. Simplification of mechanical feedback linkages within O.T. box, hence possible reduction in pitch error due to backlash.

Elimination of spool valve offset error as in (b).

Hand control of pitch position still possible.

(d) Servomotor position feedback using Electrical measurement of achieved pitch.



Disadvantages. Does not provide a mechanical pitch indicator without addition of mechanical linkage which is not essential for remote control.

Local emergency operation will be limited to a rate control with which it will be difficult to maintain a steady pitch.

Increase in number of control system and hydraulic pitch setting system interfaces.

Maintainability degraded by need to insert transducers into O.T. box.

Advantages. Eliminates complex mechanical feedback linkages and their inherent backlash.

Permits greater freedom of orientation of actuator and control valve.

Offset of spool valve when holding pitch is compensated improving system accuracy.

After careful consideration, alternative (a) has been selected as most appropriate for future warships, principally on the grounds of ease of usage in service.

In selecting the feedback system which gives greatest convenience to ships and base staffs, we appear to be accepting a possible error of up to $.3^{\circ} \pm$ from pitch demanded. This would, of course, be quite deplorable. It should however be noted that in practice a number of factors supervene.

Fortunately the control valve design ensures that a small constant load is applied to the actuator, thereby eliminating gearing backlash except during transient conditions. The small load referred to is a consequence of the control valve spool being connected to its piston rod at one end of the spool only. An unbalanced force results, due to the cross sectional area of the piston rod.

The $.18^{\circ}$ control valve deadband is a function of centrifugal force on the blades and is therefore less at reduced powers (much less at cruising speeds). At high speeds differences in water flow as blades pass the A frame arms and the hull tends to set up pitch changing couples which in practice reduce the $.18^{\circ}$. It is doubtful whether it exceeds $.1^{\circ}$ in any circumstances met afloat - which affords a rare instance of operational forces actually helping the engineer.

We are therefore able to reply on total pitch errors of less than $\pm .2^{\circ}$ in practice.

Interface with O.T. Box

It is usual in Stone Vickers design to take angle data through a fork arm, on the pivot shaft of which are two potentiometers working on an effective angle of 50°. However, in the latest installations LVDT's are used instead. There is no lost motion in an LVDT. If either kind of potentiometer should become defective it can be replaced in a few minutes, but the LVDT is the easier of the two to set up. With controls having two potentiometers it would of course be possible to switch from one to the other without delay. On the other hand - depending on the reliability of the potentiometer - one can be used for pitch indication and the other for control.

It is also good practice to have a mechanical scale at the O.T. box, and to take particular care that it be accurately calibrated, for obvious reasons. If hand control is to be exercised at the O.T. box, then the artificer concerned needs to be able to be exact both at design pitch and at zero thrust. The amount of force required to operate an open centre control valve big enough to control the propellers of a destroyer (or an aircraft carrier, for that matter) can readily be applied by one man if the valve is accurately made. It is desirable that the mechanical hunting gear referred to be fitted rather than an electrical hunting gear to reduce the difficulties of diagnosis and checking out of controls faults, and to a lesser degree to facilitate hand control.

It has already been indicated that there is no need to combine pitch and engine control onto one lever, but nevertheless it is common practice to do so, with a view to reducing the possibility of error in obeying telegraph orders from the bridge and also, perhaps, to allow men with a lesser standard of training to be used as throttle watchkeepers.

Optimising Performance

It has already been said that the controls program is dictated largely by engine considerations at the beginning of major manoeuvres. Full scale investigation in British Destroyers of the Type 41 class indicated however that there was room for improvement in the original controls programs to achieve optimum acceleration and retardation from two related viewpoints.

Firstly, available engine power was not necessarily as fully utilised as cavitation would allow.

Secondly, the angle of attack was allowed to become excessive over part of the manoeuvre, hence not achieving optimum thrusts.

There are two possible ways of minimising this problem in future design. The first where friendly navies are concerned is by way of consultation with the British Royal Navy in respect of the very extensive investigations referred to. The second is to plan the implementation of the pitch program such that minor alterations to it can be made on first of class trials to optimise the ship/engine/propeller performance.

This is allowed for in our standard CPP pitch controls, which are based on analogue computer techniques using integrated circuits.

The relationship between engine speed and propeller pitch is produced by a pair of identical variable diode function generator circuits mounted on a single P.C.B. For ease of setting up, the relationship is a series of straight lines with slopes, breakpoints and limits adjusted by means of 7 potentiometers for each curve. Figure 7 illustrates the effect of each adjustment and Figure 8 shows a typical pitch/engine speed relationship. To ensure the pitch and engine speed alter in unison irrespective of how fast the command lever is moved, a rate circuit is incorporated between the input signal and the function generator. Movement of the control lever from full ahead to full astern thus results in a dwell at zero pitch which improves stopping distance of the vessel.

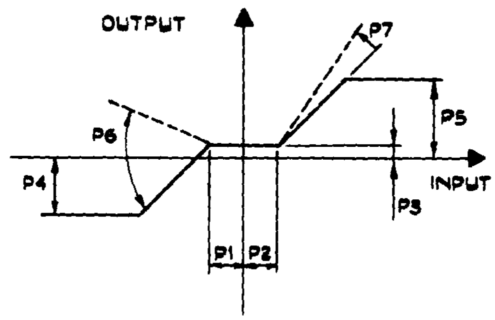


Figure 7. Profiler Characteristics.

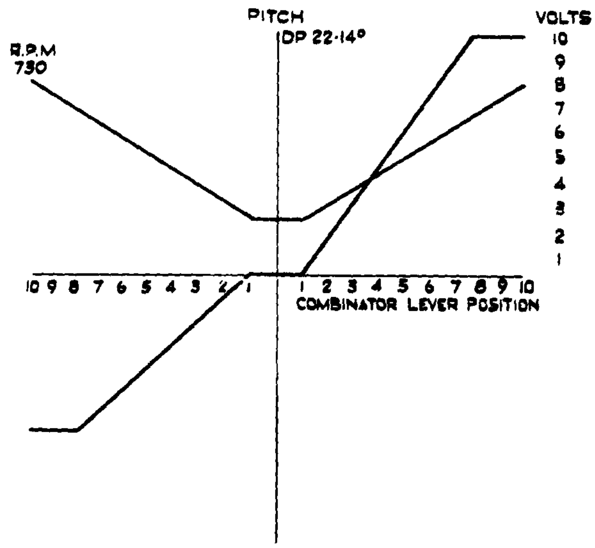


Figure 8. Combinator Diagram.

CONTROL REQUIREMENTS FOR FUTURE CPPs

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ABSTRACT

The requirement for more precise and accurate control of CPP pitch position and pitch changes is of paramount importance in ship performance and minimization of propeller noise. Very few control engineers have an understanding of the forces, hydromechanics and the engineering involved with high power controllable pitch propellers, whilst conversely very few marine engineers have a complete understanding of the present generation of functions and circuitry associated with the controlling elements.

The aim of this paper is to firstly outline the mechanics and hydromechanics of a CPP and their effect on ship performance, and then to develop that theme with respect to the control of pitch position and changes of pitch. Various examples of CPP systems will be used to illustrate the present state of the art. The paper will then continue to develop the requirements of the controlling mechanism and associated controls, outlining where and possibly how improvements can be made and where improvements will have to be made to meet the more stringent pitch accuracy and pitch repeatability required for the minimization of propeller noise.

INTRODUCTION

Nearly all major gas-turbine powered warships up to 10000 tons, designed and built in the last ten years, have been fitted with controllable pitch propellers. This has been brought about, not because CPPs offer any advantage in propulsion efficiency, but because of the unidirectional output of gas-turbine engines, the aversion at this size of warship to the use of reversing gearboxes, and the substantial manouvering capabilities provided.

At Table 1 is a list of the disadvantages of CPPs when viewed from the position of propulsion efficiency:

TABLE 1

DISADVANTAGES OF CPPs IN TERMS OF PROPULSION EFFICIENCY

1. Thrust/unit area for same diameter propeller has to be greater for CPPs. Approximately 10% of thrust.

2. Larger hubs and thicker blade root sections degrade cavitation performance.

3. Variations in 1 and 2 can lead to an earlier thrust breakdown.

Notwithstanding these disadvantages, the CPP can be viewed as a propulsive device which, because of its controllable nature has certain characteristics which can be employed to advantage. This is particularly true in the field of controlling noise emission. The concept of using CPPs to make the best of this advantage is not a new consideration and has been proposed by various naval authorities over the last six years. It is only recently, that, by collating results from various trials, and especially from the results of trials on the Canadian DDH 280 class, the picture has become clearer. These results indicate the degree of design detail of mechanical components and control systems that are now required.

The actuating mechanics of moving the blades of the propeller hub and, controlling ship manoeuvering, has been well established and is achieved by hydraulically driven mechanical linkages to the blades. A simplified view of a typical arrangement is at Fig 1. This type of design has proven to be highly reliable when it is considered that the environment in which they operate is 'hostile'.

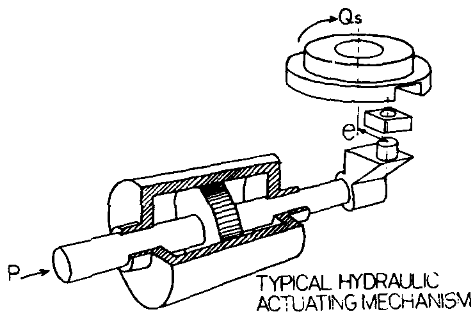


Fig 1

The control systems currently available are reliable, relatively simple and easy to maintain, and are capable of changing and holding the position of the blades within reasonably small errors. However the evidence gathered from trials has indicated that these small control system errors, along with errors in the hub and blades, can influence the inception speeds of the various types of cavitation. These in turn, influence the noise characteristics of the propellers particularly between 10 and 18 knots. Table 2 indicates the maximum errors that can be encountered.

TABLE 2

TYPICAL MAXIMUM ERRORS IN DESIGN PITCH

1. Discrepancies between identically designed and manufactured blades - 0.25 degrees.
2. Pitch repeatability for a given pitch setting - 0.5 degrees.
3. Pitch holding accuracies - 0.5 degrees.
4. Wear and clearances in the blade linkages in the hub - 0.25 degrees.

In spite of the existance of these errors and the need to reduce them, trials have also indicated that by increasing pitch above the design setting, by some three degrees, a decrease in the noise emission characteristic is achieved. Typically propeller emitted noise can be reduced by up to 4 db at 100 Hz and 8db, at 10 KHz.

Because a CPP can be described as an "Infinite number of propellers, where each pitch setting is a new propeller on the same hull form"[1] then within this context, the pursuit of noise reduction offers a challenge, not only to the propeller designer but to the control systems designer. To illustrate the control requirements for future CPPs this paper outlines the propeller forces involved, problems of measuring pitch accurately and methods of controlling the pitch. This is then developed into some ideas as to how the goal of minimum noise can be attained.

CONTROL OF CPPs

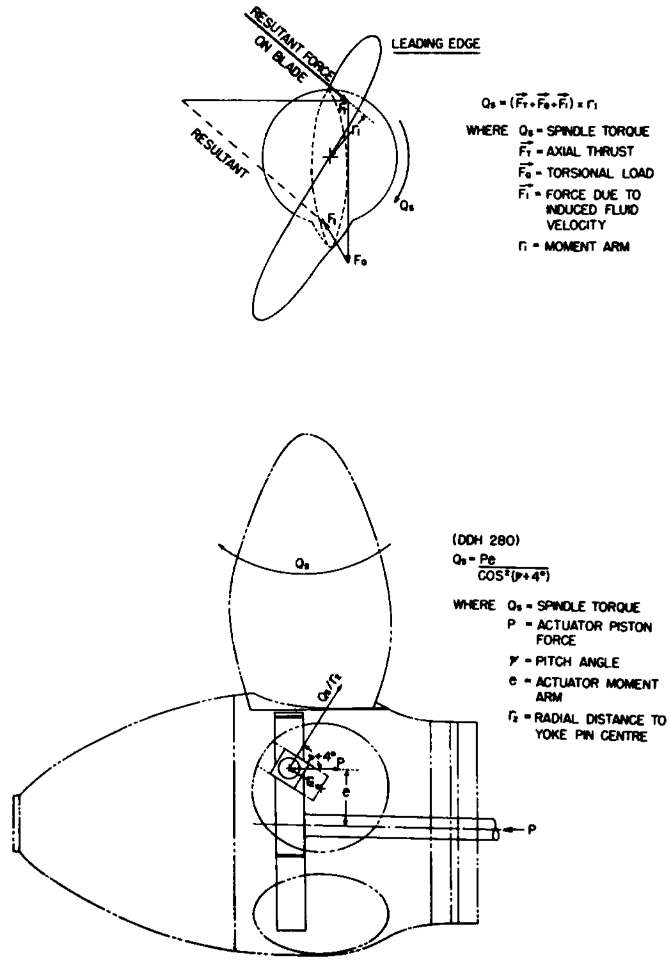
Mechanics of CPPs

The modern controllable pitch propeller mechanism employs the operating principle of converting a linear applied force into torque. Because of the constraints on hub size and the difficulties presented by the magnitude of the forces involved, it is currently impractical to use a mechanism which applies a direct torque to the blades. The DDH 280 actuating mechanism is typical 'slot-pin' mechanism, and uses an actuator cylinder located inside the ship, remote from the propeller hub. It is primarily the experience with this equipment that guides this discussion of propeller mechanics.

The wide variety of conditions under which it is possible to change the pitch of a CPP makes it possible to introduce an equally wide range of forces; and it becomes of great importance to balance the strength requirements of the mechanism with the control requirements imposed upon it. Large actuating forces may be required to introduce a pitch change or maintain conditions of constant desired pitch. The frictional effects of these large actuating mechanisms, the expectation of frequent high shock loadings and the increasingly strict requirements on pitch accuracy and repeatability underlines the importance of this balance.

The hydrodynamic force acting on a propeller developing thrust depends primarily on propeller pitch and shaft RPM. The actuator force required to overcome this hydrodynamic force depends on the blade shape in as much as its line of action determines the spindle torque to be overcome by the actuator. In the DDH 280, the capability to overcome this spindle torque is measured by the maximum actuator force 'P' and the moment arm 'e' in figure 2. It is important to realize that recent trends in new design propellers tend to result in highly skewed or irregularly shaped blades. This is likely to cause increased or irregular torque effects over the propeller's pitch range owing to the probable variations in the force line of action. It should therefore be expected that spindle torques will not decrease in new designs.

Control of the pitch/RPM schedule during normal operation can significantly reduce the magnitude of the required actuating forces. An envelope of actuating force, figure 3, defines the worst conditions which could be experienced in full power operation of the DDH 280. (Curves for other CPPs are similar in shape) Operationally determined minimums of performance generally define the most favourable conditions. A pitch/RPM characteristic, designed into the control of the actuator mechanism, will establish an operating evelope with much lower peak actuation forces and will hopefully produce an optimum balance between the two constraints. The DDH 280 operating line from zero thrust to full power ahead is illustrated in figure 3 as an example.



RESULTANT FORCE ON BLADE
LEADING EDGE
RESULTANT
$Q_s = (\vec{F_T} + \vec{F_\theta} + \vec{F_I}) \times r_1$
WHERE Q_s = SPINDLE TORQUE
$\vec{F_T}$ = AXIAL THRUST
$\vec{F_\theta}$ = TORSIONAL LOAD
$\vec{F_I}$ = FORCE DUE TO INDUCED FLUID VELOCITY
r_1 = MOMENT ARM
F_θ
Q_s
(DDH 280)
$Q_s = \frac{Pe}{\cos^2(\gamma + 4°)}$
WHERE Q_s = SPINDLE TORQUE
P = ACTUATOR PISTON FORCE
γ = PITCH ANGLE
e = ACTUATOR MOMENT ARM
r_2 = RADIAL DISTANCE TO YOKE PIN CENTRE
Q_s/r_2
$\gamma+4°$
P
e

Fig 2

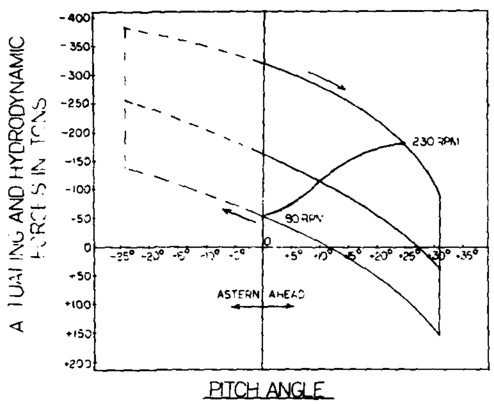


Fig 3

The force effect of friction in a CPP actuator mechanism represents a large percentage of the total actuator force which must be developed to achieve a pitch change. The vertical distance between the calculated actuator forces and the hydro dynamic forces (middle line) in figure is representative of the magnitude of this effect. From a control point of view, it is of some advantage that this effect is relatively constant and independent of the propeller ouput; and disadvantageous in its magnitude.

The designer of the pitch actuator may take account of the large external influences on the propeller pitch by designing the entire system for a measure of insensitivity to these shock loadings. Alternatively, he may design in, a pitch locking arrangement which, when engaged, makes the system insensitive to shock loads maintaining pitch once the set point has been achieved. If it is considered preferrable to increase the capacity of the pitch actuator to enable continuous control to be achieved, the mechanical components required for this application will likely be approaching the limits of satisfactory operation due to the large operating range of forces necessitated as a result. However, if a locking arrangement is employed with its attendant problems of deadband, the mechanical components may still be overtaxed and the required limits on control may not be achieved. The overall consideration here should be one of selection of control system hardware.

Measurement of Pitch

In almost all controllable pitch propeller designs, the actual pitch of the propeller is measured by mechanical linkages connecting the hub/blade mechanisms, of the shaft into the ship, to the pitch control system, and incorporates a device which transmits a rotating longitudinal movement to a non rotating longitudinal movement. Therefore, depending upon the position of the oil transfer block, this device is either at the end of the shaft, (through the main wheel in the gearbox), fig 4, or at a position between the gearbox and the shaft stern seal fig. 5. The latter method, by its physical nature introduces a greater error in the actual pitch measurement but dependent upon the detail mechanical design, accuracies of better than ±0.2 degrees of pitch are theoretically possible.

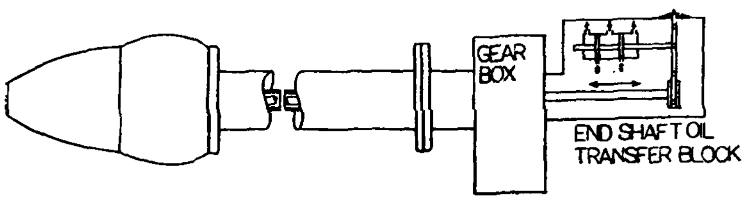


Fig 4

Other types of pitch measurement have been employed or are being considered and linear variable differential transformers (LVDT) have been used in the hub. These types of measuring devices have so far been restricted to trials purposes for measurement only and have to date not been employed as part of the control system. Reasonably accurate measurement was achieved, typically 0.1 of a degree but the long term reliability in a warship environment was never evaluated.

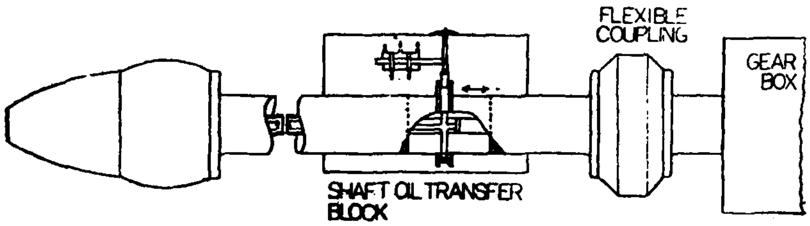


Fig 5

It is possible to conceive of many ways of actually measuring the pitch of the blades accurately, and transmitting this measurement to the control system some 100 *ft* up the shaft into the ship. In the writing of this paper, much thought was given to this particular problem, but it was concluded that in weighing the advantages of a small increase in accuracy of pitch measurement against reliability and greater complexity the present designs of mechanical pitch measurement give the best solution. This is not to say however, that other means of pitch measurement should not be investigated.

CPP Hydraulics

A typical servosystem can be simply illustrated as in fig 6:

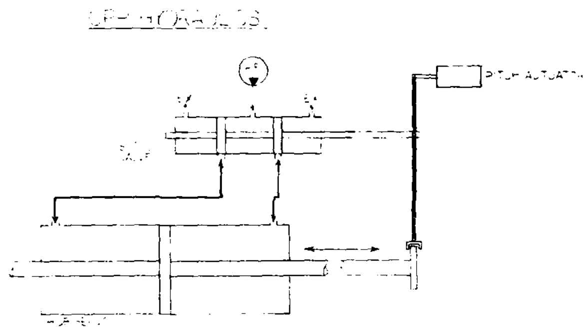


Fig 6

The pitch actuation can be pneumatic, hydraulic, electrical or even mechanical cable in simple systems. To fit in with present ship controls the pitch actuator is usually an electrical device and typically a stepper motor. The pump has to be capable of not only generating the pressures to overcome the actuating forces but the volume required, and are usually of the positive displacement type. Typical ratings are up to 1000 psi and 100 galls/min but the output is usually selected for a given system to limit the pitch change rate.

The spool valve represents perhaps the component, that requires the most design effort and precision, because in this typical type of system its characteristics should be matched as far as possible to the requirements for actuating and holding pitch.

The above statement implies that when the highest pitch accuracy and pitch holding is required, i.e, under steady steaming conditions, then the overall hydraulic system sensitivity should be the highest. Unfortunately this cannot be accomplished easily with a conventional type of spool valve because of two conflicting requirements. These being:

a. A large movement of the spool valve is required to change pitch when the actuating forces are high.

b. A small movement of the spool valve is required to hold at ahead design pitch when the actuating forces can be a small percentage of the pitch change actuating forces (usually less than 30%).

A simple relationship between hub actuating pressure and the blade spindle torque is:

$$P = \left[\frac{Qs\ \cos^2\psi}{Ae}\right] + PB$$

where Qs = is the spindle torque for all blades

ψ = pitch angle

A = hub sero piston area

e = slide block drive pin distance from the centre line

P = the hub actuating pressure

PB = hub return oil pressure

From this expression, it can be seen that the control of the spool valve position is critical, not only in terms of the oil pressure delivered to the hub, but the control of the return oil from the hub. Fig. 7 shows the flow curves for a single spool valve with under lapping

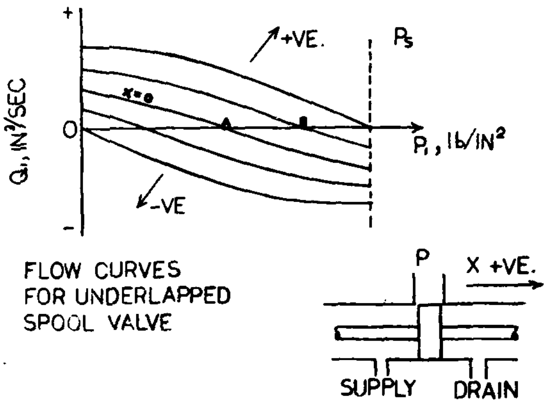


Fig 7

which amplifies the fact that for larger values of P, the spool valve opening is increased for a zero flow, i.e., point B instead of point A, and the only way to realize a larger P for a constant spindle torque is to increase the value of PB. This illustrates only one small aspect of the development work that has to go into spool valve design, and it is acknowledged that certain major CPP manufacturers devote design and development effort to this field. However it is considered that a large amount of further work can be undertaken that would realize better pitch holding accuracies.

FUTURE REQUIREMENTS

The two major aspects that are required for future CPPs that have been previously discussed can be itemized as:

a. The pursuit of greater pitch accuracy through better designs of hydraulic circuitry and pitch measurement.

b. Control of the pitch over a small range (within ±5 degrees around the full power design pitch) to achieve noise reduction in the lower speed ranges.

Some outlines on how to achieve the first aspect and some of the avenues open for investigation have been addressed in this paper. However it is the second aspect which it is felt will realize the best returns on noise reduction for the efforts employed.

It is envisaged that the contral system associated with a CPP design, which aims to reduce noise by selecting the optimum pitch for a given ship speed, will require the pitch control to be integrated with the shaft speed and torque. This will enable the propulsive thrust to be matched to a desired ship speed. In essence, the control of the ship's speed would be based on thrust, with shaft speed and pitch being the variables controlled for minimum noise. From the trials conducted to date it is postulated that the revised pitch/shaft RPM schedule would differ from the present type of pitch/shaft RPM schedule as shown in fig 8.

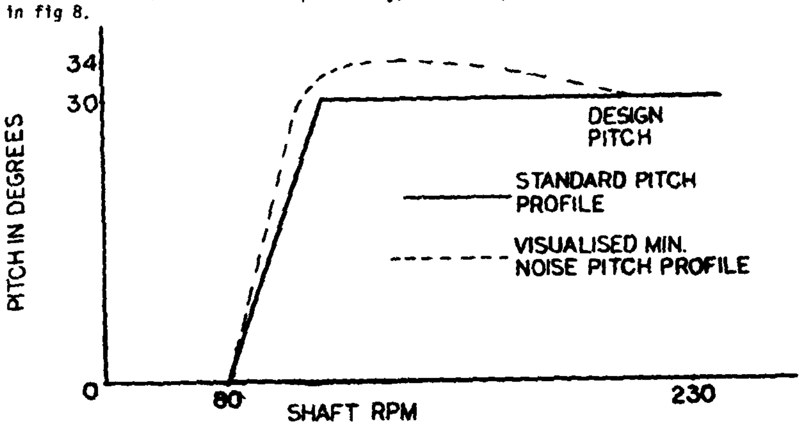
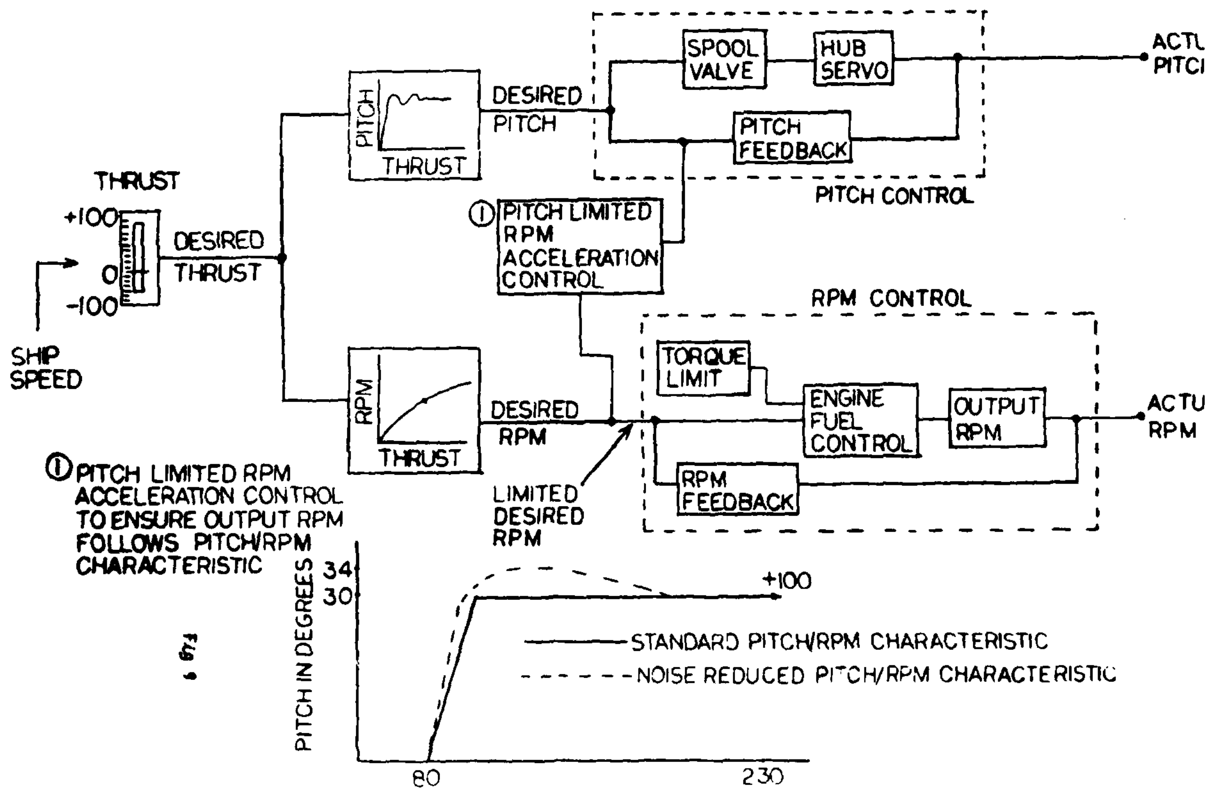


Fig 8

The actual 'shape' of this minimum noise schedule would obviously depend upon the design of the propeller and at what ship speed/propeller RPM each form of blade cavitation becomes dominant. For example at low ship speed blade tip cavitation is dominant over hub vortex cavitation whilst at higher ship speeds bubble cavitation, with its lower frequency noise may be less desirable than face sheet cavitation with its higher frequency noise. At fig 9 is a basic outline of a control schematic which for a given demand for ship speed, a percentage thrust is set, which in turn sets the two variables, propeller RPM and pitch, to give the minimum noise.

It is conceivable that the 'loop' on the whole pitch/RPM control can be closed by feeding back a noise signal from a hydrophone mounted on the hull near the propeller. So far this idea has been discounted because of the complexity of the noise emitted from a propeller and it would be difficult, if not impossible, to filter out stray hull emitted noise. The concept of adjusting the pitch/RPM schedule for minimum noise has some justification for being the course of action likely to produce a significant positive overall effect on noise emission - these schedules being set by trials in the first of class.


THRUST
+100
0
-100
SHIP SPEED
DESIRED THRUST
PITCH
THRUST
DESIRED PITCH
SPOOL VALVE
HUB SERVO
PITCH FEEDBACK
PITCH CONTROL
ACTU PITC
① PITCH LIMITED RPM ACCELERATION CONTROL
RPM CONTROL
RPM
THRUST
DESIRED RPM
TORQUE LIMIT
ENGINE FUEL CONTROL
OUTPUT RPM
RPM FEEDBACK
ACTU RPM
LIMITED DESIRED RPM
① PITCH LIMITED RPM ACCELERATION CONTROL TO ENSURE OUTPUT RPM FOLLOWS PITCH/RPM CHARACTERISTIC
PITCH IN DEGREES
34
30
+100
80
230
STANDARD PITCH/RPM CHARACTERISTIC
NOISE REDUCED PITCH/RPM CHARACTERISTIC

P 2-10

Fig 9

CONCLUSIONS

This paper has attempted to outline the areas of CPPs and their control systems in which improvements could be made, i.e., minimization of propeller noise and reduction of the detection/acquisition range being of prime importance. To substantially achieve this end, concentration should be given to using the modifications proposed herein. Existing warship CPP control systems are configured such that it is entirely feasible to replace the pitch/RPM schedule with one specifically designed for low noise, with little alteration to the installation. This approach should be pursued by CPP designers and manufacturers.

References:

1. Propeller Viewing and Photographic Trials: USS BARBEY (FF-1088) with a controllable pitch propeller (CONFIDENTIAL) DTNSRDC 78/035 Terry Brocket March 1978.

2. DDH 280 Propeller Viewing and Sound Ranging Trial: Design Pitch and Single Shaft Operation (CONFIDENTIAL) L.J. Leggat, J.L. Kennedy August 1980.

3. Report on HMS Birmingham Propeller Viewing Trial AMTE (H) R78038 (CONFIDENTIAL) C.B. Wills September 1978.

MULTIVARIABLE CONTROL OF A SHIP PROPULSION SYSTEM

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ABSTRACT

The application of multivariable control theory to a gas-turbine powered, variable pitch propeller propulsion system is considered. The propulsion system model, which is described by non-linear algebraic-differential equations, is linearized about various operating points and transformed into transfer function matrix form using system matrix theory. The non-linear and non-minimum phase characteristics of the propulsion plant are discussed and it is shown that an adaptive multivariable controller can be constructed from a number of linear multivariable designs which can cope with the non-linear nature of the plant, and also prevent overstraining of the propeller shaft without restricting the performance of the propulsion plant.

INTRODUCTION

The availability of low cost microprocessor systems and the significant developments over the last decade in linear multivariable control theory, are the main motivations for developing alternative control schemes for marine gas-turbine, variable pitch propeller propulsion plants, coupled with the need to prevent overstraining of the propeller shaft and possible breakage, without adversely affecting the overall performance of the propulsion machinery.

Typical control schemes in current use evolved from large scale simulation studies of the plant and employ single loop control of the shaft speed. Pitch and fuel demands are scheduled from a power lever in such a way as to give optimum plant efficiency under steady state conditions. Overstraining of the propeller shaft is prevented by the use of rate limits and saturation elements that can, under large step manoeuvre conditions, restrict the performance of the propulsion machinery. The resulting control scheme therefore utilises both closed-loop and open-loop systems with restrictive non-linear elements keeping the torque developed within acceptable, safe, operating limits.

Overstraining of the propeller shaft can, however, also be prevented by closed loop control of both the propeller shaft speed and the propeller shaft torque via manipulation of the two system inputs of gas-turbine fuel flow and propeller pitch demand. The system model now takes the form of a non-linear coupled multivariable configuration with two inputs and two outputs where, in general, each input change affects both outputs. With direct closed-loop control exercised over both shaft speed and shaft torque, the restrictive rate limits required by conventional control schemes to prevent overstraining of the shaft are no longer required and can therefore be relaxed. Consequently, the resulting control system relies on the use of two decoupled closed-loop control systems to regulate shaft speed and shaft torque, with less restrictive rate limits being used in a secondary back-up role to prevent overstraining under fault conditions, such as loop failure.

The decoupling of the multivariable system configuration can be achieved by applying linear multivariable design techniques to linearized transfer function models selected from the ship operating envelope. Developments in linear system theory over the last decade enable these linearized transfer function models to be obtained from the nonlinear algebraic-differential equations describing the plant behaviour. Earlier work [1] using turbine torque and propeller shaft speed as controlled outputs established the feasibility of using multivariable compensators to control the propulsion plant, but did not overcome the need to use restrictive rate limits to prevent overstraining the propeller shaft. The dynamic and steady state characteristics of the propulsion plant are highly non-linear, and a wide variation in the speed of response of the system is experienced over the ship operating envelope. Additionally, non-minimum phase characteristics are encountered within some regions of the operating envelope which are inherent to the basic characteristics of the components of the propulsion plant. Consequently the final control system is constructed using the linear designs such that the parameters of the controller adapt to the varying load conditions of the plant. This ensures that the desired performance is obtained in face of the non-linear and non-minimum phase characteristics of the plant. The new control scheme, being largely engineered by way of software, can be easily constructed to smoothly schedule the controller in accordance with the loads placed upon the plant, whilst at the same time maintaining strict upper torque and speed limits.

MATHEMATICAL MODEL

The ship and propulsion machinery are modelled using a FORTRAN IV digital computer simulation which includes steady-state data for the hull resistance characteristic, the propeller torque characteristic and the propeller thrust characteristic. The dynamic response of the system is non-linear and is represented by the following equations:

$$2\pi I_T \dot{N}_p = rQ_t - Q_p(N_p, V_s, \phi_a) \tag{1}$$

$$M \dot{V}_s = 2T_p(N_p, V_s, \phi_a) - R_s(V_s) \tag{2}$$

$$Q_t + \tau \dot{Q}_t = Q_D(F_d, N_t) \tag{3}$$

$$\phi_a + \tau_4 \dot{\phi}_a = \phi_d \tag{4}$$

(the nomenclature is given in Appendix 1). A block diagram of the system is given in Figure 1.

MULTIVARIABLE TRANSFER FUNCTION MODELS

Previous multivariable control system design studies by Winterbone et al [1] used propeller speed N_p and turbine torque Q_t as controlled system outputs, and the gas-turbine fuelling rate F_d and the propeller pitch angle ϕ_d as the manipulated system inputs. However, exercising closed-loop control over the turbine torque and the propeller shaft speed does not necessarily prevent overstraining of the propeller shaft and possible shaft breakage during transient conditions resulting from large step manoeuvres. Protection from overstraining under these circumstances can only be achieved by the use of additional rate limits on the system variables, which in turn can restrict the overall performance of the propulsion plant under large step manoeuvre conditions. It is therefore desirable that closed loop control be directly exercised over the shaft torque and shaft speed in order to prevent overstraining, less restrictive rate limits being used in a secondary back up role to prevent overstraining under fault conditions such as loop failure. Hence the linear multivariable model is defined in transfer function form as:

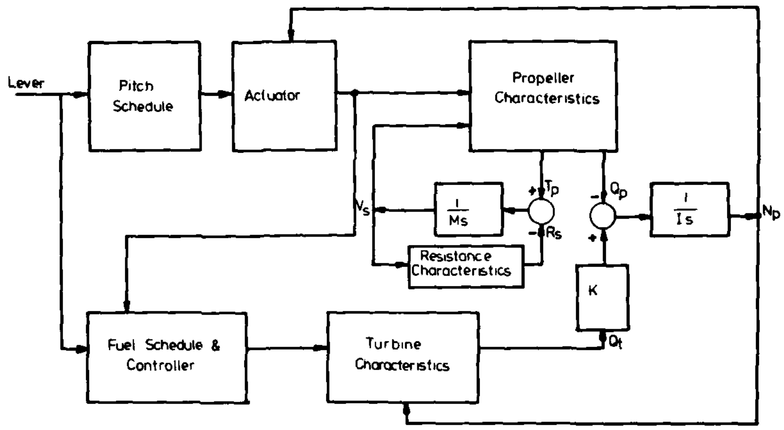

Figure 1. Schematic of Propulsion System.

$$\begin{pmatrix} N_p(s) \\ Q_s(s) \end{pmatrix} = \begin{pmatrix} g_{11}(s) & g_{12}(s) \\ g_{21}(s) & g_{22}(s) \end{pmatrix} \begin{pmatrix} F_d(s) \\ {}_d(s) \end{pmatrix} \tag{5}$$

Revised System Model

With the available mathematical model of the propulsion plant, one of the desired outputs, the shaft torque Q_s, is not readily available and some modifications to the plant model are thus necessary. The existing plant model as described by equations (1) to (4) makes no allowance for the stiffness associated with the propeller shaft. The shaft torque can be obtained from a mathematical model which includes this shaft stiffness, and the resulting state space description of the plant would have six state variables. However, this approach to the problem is made intractable by the difficulties of assigning a value to the propeller shaft stiffness.

To overcome this problem an alternative approach was adopted that avoided using the shaft stiffness. The propeller shaft was considered to be a rigid body, the moment of inertia of which could be lumped with the moment of inertia of the propeller. Under this assumption it is possible to write down a torque balance for the propeller shaft. Thus, the shaft torque Q_s is given as

$$Q_s = 2\pi I_p \dot{N}_p + Q_p(N_p, V_s, \phi_a) \tag{6}$$

The above equation when added to equations (1) to (4) comprises a set of algebraic differential equations that cannot now be readily rearranged into a state space form. If a space ν spanned by N_p, V_s, Q_t, Q_s and ϕ_a is defined, then the four-dimensional state space defined by equations (1) to (4) is a four-dimensional subspace of ν. In this situation the normal transformation from the state-space form to transfer function form defined by:

$$G(s) = C'(sI-A)^{-1}B + D \tag{7}$$

cannot be used. However, on linearizing equations (1) to (4) and equation (6), and taking Laplace transforms, the above equations can be rearranged into the following general form, given by:

$$\begin{aligned} T(s)\bar{\xi} &= U(s)\,\bar{u} \\ \bar{y} &= V(s)\bar{\xi} + W(s)\,\bar{u} \end{aligned} \tag{8}$$

and thus placed into the system matrix form as defined by Rosenbrock [2], given as:

$$P(s)\begin{bmatrix} \bar{\xi} \\ -\bar{u} \end{bmatrix} = \begin{bmatrix} \underline{0} \\ -\bar{y} \end{bmatrix} \tag{9}$$

where $P(s)$, the system matrix is given as:

$$P(s) = \left[\begin{array}{c|c} T(s) & U(s) \\ \hline -V(s) & W(s) \end{array}\right] \tag{10}$$

and where the vector of Laplace transformed system variables $\bar{\xi}$, the vector of Laplace transformed input variables $\bar{u}$ and the vector of Laplace transformed output variables $\bar{y}$ are given as:

$$\bar{\xi} = \begin{bmatrix} V_s \\ Q_t \\ N_p \\ Q_s \\ \phi_a \end{bmatrix}, \quad \bar{u} = \begin{bmatrix} F_d \\ \phi_d \end{bmatrix}, \quad \bar{y} = \begin{bmatrix} N_p \\ Q_s \end{bmatrix} \tag{11}$$

The linearized transfer function matrices for various operating points within the ship operating envelope are now obtained from the relationship:

$$G(s) = V(s)T^{-1}(s)U(s) + W(s) \tag{12}$$

which is a generalization of equation (7).

Results of Linearization

The system matrix $P(s)$ is composed of partial derivatives which can be evaluated analytically from the FORTRAN IV simulation model. The parametric form of the transfer function matrix which results from the linearization and the transformation from P(s) to G(s) is given as:

$$G(s) = \frac{1}{d(s)} \begin{pmatrix} k_{11}(s + 1/T_2) & \dfrac{k_{12}(s + 1/T\ )(s + 1/T_1)}{(s + 1/\tau_4)} \\ k_{21}(s + 1/T_3)(s + 1/T_4) & \dfrac{k_{22}(s + 1/T_1)\ n(s)}{(s + 1/\tau_4)} \end{pmatrix} \tag{13}$$

giving

$$\begin{pmatrix} N_p(s) \\ Q_s(s) \end{pmatrix} = G(s) \begin{pmatrix} F_d(s) \\ \phi_d(s) \end{pmatrix} \tag{14}$$

where $d(s) = (s + 1/\tau_1)(s + 1/\tau_2)(s + 1/\tau_3)$ in the case of all real poles, and

$$d(s) = (s + 1/\tau_1)(s^2 + 2\sigma_2 s + \sigma_2^2 + \omega_2^2)$$

for the case of a pair of complex conjugate poles. The polynomial $n(s)$ is given as $n(s) = (s + 1/T_5)(s + 1/T_6)$ for the case of two real zeros, and

$$n(s) = (s^2 + 2\sigma_1 s + \sigma_1^2 + \omega_1^2)$$

for the case of two complex conjugate zeros.

Table 1 is an abridged table of normalized steady-state gains over the ship operating envelope. Tables 2 and 3 are abridged tables of the dynamics at the same operating points. These show that the system is extremely non-linear with variations in both gains and dynamic terms of the order of a hundred to one.

Table 1. Steady state gain variation over operating range

Fuel %	Pitch %	$\frac{k_{12}}{(k_{12})_{max}}$	$\frac{k_{11}}{(k_{11})_{max}}$	$\frac{k_{22}}{(k_{22})_{max}}$	$\frac{k_{21}}{(k_{21})_{max}}$
95	95	1.00	0.127	1.000	0.842
40	95	0.337	0.246	0.212	0.933
5	30	0.907	1.000	0.351	0.511
10	-80	0.601	0.501	0.252	1.000

Table 2. Variation of system poles over operating range
(* all poles real $\tau_2/(\tau_2)_{max}$ and $\tau_3/(\tau_3)_{max}$)

Fuel %	Pitch %	$\frac{\tau_1}{(\tau_1)_{max}}$	$\frac{\sigma_2}{(\sigma_2)_{max}}$	$\frac{\omega_2}{(\omega_2)_{max}}$	$\frac{\tau_4}{(\tau_4)_{max}}$
95	95	0.110	1.000	0.893	0.535
40	95	0.226	0.802	1.000	0.651
5	30	0.376	*0.900	0.897	0.868
10	-80	0.502	0.579	0.848	0.899

Table 3. Variation of system zeros over operating range
(† zeros real $T_5/(T_5)_{max}$ and $T_6/(T_6)_{max}$)

Fuel %	Pitch %	$\frac{T_1}{(T_1)_{max}}$	$\frac{T_2}{(T_2)_{max}}$	$\frac{T_3}{(T_3)_{max}}$	$\frac{T_4}{(T_4)_{max}}$	$\frac{\sigma_1}{(\sigma_1)_{max}}$	$\frac{\omega_1}{(\omega_1)_{max}}$
95	95	0.005	0.124	0.095	0.057	1.000	1.000
40	95	0.023	0.215	0.201	0.081	1.000	0.527
5	30	0.012	0.547	0.382	0.813	†1.000	0.854
10	-80	0.018	0.500	0.463	0.157	†0.842	0.938

At small pitch angles (between -15% and +30%) all the system poles are real but for higher loads two of these become a pair of complex conjugate poles that move leftward in the s-plane with increasing pitch angle. The basic response of the system is such that it becomes faster at higher loads than at idling. In all cases the dominant pole is real and has an associated time constant which varies by a factor of six between 5% load and 90% load.

Non-Minimum Phase Behaviour

The element zero ($= -1/T_1$) common to elements $g_{12}(s)$ and $g_{22}(s)$ of $G(s)$ moves into the right half s-plane for pitch angles at the extrema of the pitch angle range, thus causing the propulsion plant to display non-minimum phase characteristics at these extreme pitch angles. Analysis of the general expressions for the polynomials in equation (13) and of the partial derivatives obtained from the linearization of the non-linear dynamic equations by Kidd [3] have indicated that this non-minimum phase behaviour may be due to excessive thrust loading resulting from changes in pitch angle inducing cavitation on the back of the propeller blades, thus reducing the thrust developed by the propeller for a given pitch angle and propeller speed. This effect may be negated at higher ship speeds by the increasing ship resistance which results in a change in the speed of advance of the propeller through the water. Consequently non-minimum phase characteristics appear to be an inherent characteristic of this kind of propulsion plant.

Time Responses

Figures 2 and 3 show time responses of normalized outputs to step changes in normalized inputs. The parameters were normalized by dividing by the maximum values, hence the outputs

$$\text{Normalized Speed} = \frac{N_p}{N_{p_{max}}}$$

$$\text{Normalized Torque} = \frac{Q_s}{Q_{s_{max}}}$$

and the inputs are:

$$\text{Normalized Fuel} = \frac{F_d}{F_{d_{max}}}$$

$$\text{Normalized Pitch} = \frac{\phi_d}{\phi_{d_{max}}}$$

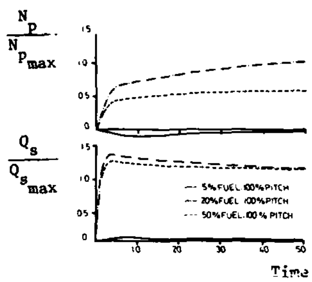

Figure 2. Response of Open-Loop System to Step Input on Fuel.

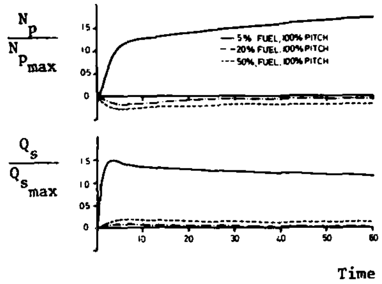

Figure 3. Response of Open-Loop System to Step Input on Pitch.

The diagrams show that at different operating points the response rate changes; the system responds more rapidly at high loads than low ones. It is also apparent that interaction between input 2 and output 1 (i.e. ϕ_d and N_p) becomes greater at higher loads, whereas the other cross-coupling gain is only a weak function of load level.

MULTIVARIABLE COMPENSATOR DESIGNS

The philosophy of the UMIST design approach consists of two distinct phases. In the first phase, linear transfer function models selected from various points within the ship operating envelope are used to produce the structure and initial parameter values of multivariable compensators, using linear multivariable design techniques. In the second phase, the structure and parameter values of these compensators are examined, and an adaptive multivariable compensator is formed using some of the linear designs. The adaptive multivariable compensator is then implemented on the non-linear simulation model, and the parameter values tuned to take account of the non-linear and non-minimum characteristics of the plant.

Linear Compensator Designs

The design of the multivariable compensators was performed using Rosenbrock's Inverse Nyquist Array technique (INA). This method, which is fully described in [4], is based on the fact that when the inverse of the transfer function matrix is 'diagonal dominant' (i.e. very nearly diagonal) then certain theorems can be used to assess the stability and performance of the resulting closed loop multivariable system. For this design exercise the UMIST CAD package [5] was used. This enables dominance to be tested graphically by plotting the inverse of the transfer function matrix, together with Gershgorin circles, on a visual display unit (VDU). A typical example of an inverse Nyquist plot for the basic system $G^{-1}(s)$ is shown in Figure 4. This is based on 5% fuel and 50% pitch and shows that the system is not diagonally dominant because the envelope generated by the circles superimposed on the diagonal elements do not exclude the origin.

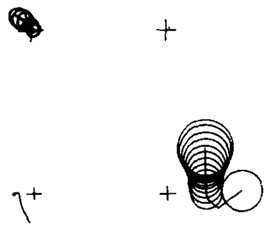

Figure 4. Inverse Nyquist Array at 5% Fuel, 50% Pitch.

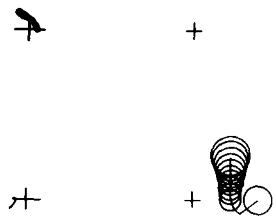

Figure 5. *Compensated Inverse Nyquist Array.*

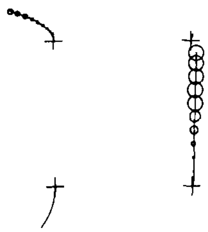

Figure 6. Closed-Loop Inverse Nyquist Array.

However, it is possible to choose a constant multivariable precompensator to make the system diagonal dominant as shown in Figure 5. Since the Gershgorin bands do not cut the negative real axis the system will be stable for arbitrarily large feedback gains (at least in theory). Even so, it is impossible to obtain satisfactory performance with this simple scheme. Since the low frequency part of the element 2 is far away from the origin the corresponding gain must be set at a sufficiently high value to reduce steady state error to an acceptable level. Thus, further single-loop compensation involving dynamic elements is necessary to achieve good closed-loop performance. It was therefore decided to employ integral action to eliminate steady state error and reshape the low frequency part of the plot. The resulting INA of the closed-loop system with feedback gains chosen to give the desired closed-loop performance is shown in Figure 6.

Implementation of an Adaptive Multivariable Controller

It was shown in the previous section that the system could be made diagonally dominant and stable compensators could be designed for small changes in input parameters. It was also noted that a single compensator could not be implemented over the whole of the ship operating envelope due to the large variation in plant parameters. An examination of the structure and parameters of the multivariable compensators obtained from the linear design exercise reveals that the adaptive multi variable controller needs to be scheduled to have different parameters within four regions of the operating envelope. The compensators designed above were based on the basic 2 input 2 output system in which either input or output may be varied independently. However, it is not desirable to make both of these inputs available to the operator; they must be amalgamated in some way and related to the propulsion power lever position. This can be achieved by scheduling shaft speed demand and shaft torque demand from power lever position in such a manner that optimum conditions are obtained in the steady state [1].

When operating over a large change of demand the propulsion unit will require the parameters of the controller to be modified to suit the actual region of operation, i.e. the controller will adapt to the state of the system. This adaptation can most easily be brought about by relating the controller parameters to the propeller speed. It is also necessary to incorporate some further simple logic to accommodate the conditions which occur during direction change manoeuvres, such as a full ahead to full astern manoeuvre. The simple logic assures that the signal applied to the pitch actuator control system is in the correct sense during direction change manoeuvres. Figure 7 shows a schematic of the overall control system. This shows the compensator matrix which has been incorporated to achieve diagonal dominance and also shows the additional propeller speed and pitch signals which are required to schedule the controller and cope with changes of direction of travel respectively.

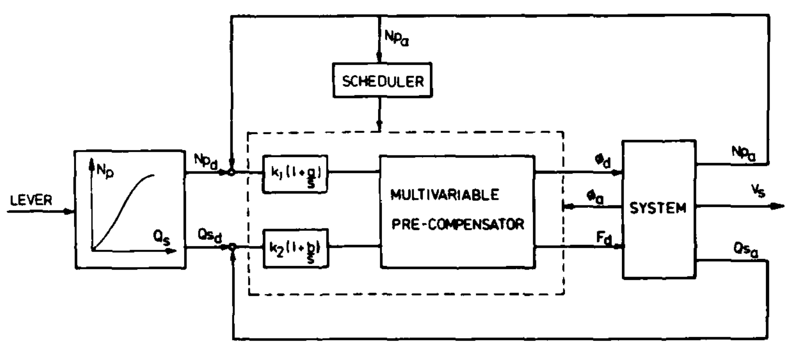

Figure 7. Multivariable Control Scheme.

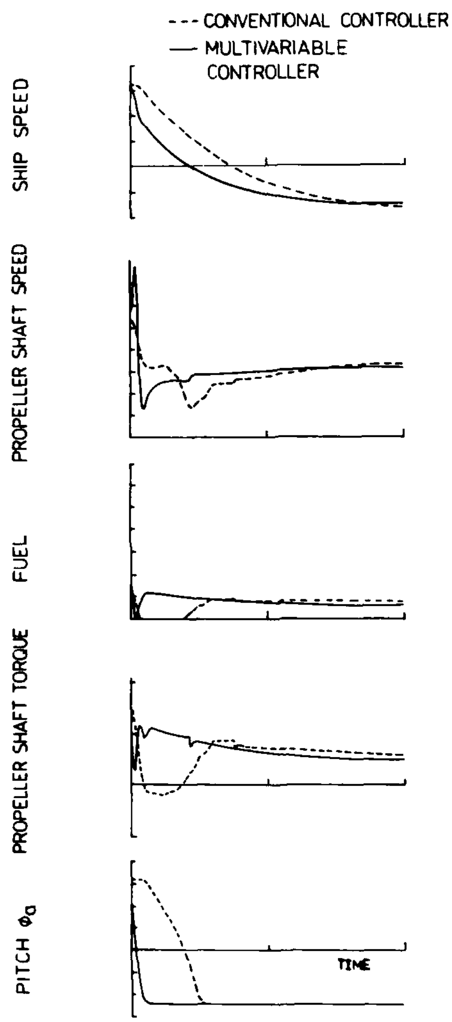

Figure 8. Full Ahead to Full Astern Manoeuvre.

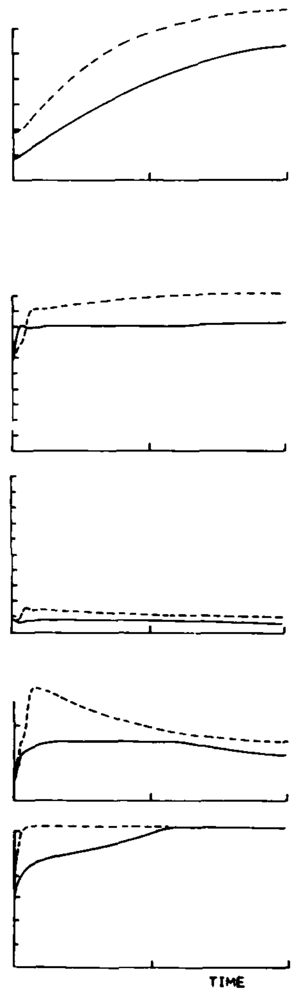

Figure 9. Small Step on Lever Demand: 16% to 32%.

Figure 8 shows the performance of the vessel undergoing a manoeuvre from full ahead to full astern, while Figure 9 shows the performance of the vessel when the demand is changed from 16% lever position to 32% lever position. The responses obtained from the multivariable control system are those of the propulsion system with all restrictive rate limits removed, and are compared with responses obtained from a conventional control system employing these rate limits. Figure 8 indicates that a dramatic increase in the speed of response of the vessel is obtained without overstraining the propeller shaft and without using any form of predictive control strategy. These figures illustrate the best speed of response that could be obtained from the propulsion plant. The responses shown in Figure 9 show that the multivariable control system effectively controls propeller shaft speed and propeller shaft torque, and produces a vessel performance which is again different from the response obtained with a conventional control system. In this case, the results indicate that there is a difference between the propeller torque/speed schedules of the two control systems.

CONCLUSIONS

To conclude, the multivariable control system with restrictive rate limits removed is able to maintain the shaft torque within acceptable safe operating limits and the results presented indicate the potential improvement that could be obtained from a vessel utilising an adaptive multivariable controller.

Further work will be aimed at introducing relaxed rate limits and re-evaluation of the schedule used in the multivariable controller. Effects of loop failure on the performance of the system will also be evaluated, and the digital implementation of the proposed control scheme will be developed.

ACKNOWLEDGEMENT

The authors wish to thank the Ministry of Defence (P.E.) for permission to publish this paper and for financially supporting this project.

REFERENCES

[1] D.E. Winterbone, R. Whalley, C. Thiruarooran, N. Munro, "Adaptive Multivariable Control of Ship Propulsion Plant", 3rd International Symposium on Ship Operation Automation, Tokyo, Japan, 1979.

[2] H.H. Rosenbrock, "State Space and Multivariable Theory", Nelson, 1970.

[3] P.T. Kidd, "Control of a Ship Propulsion System", M.Sc. Dissertation, University of Manchester, U.K., 1980.

[4] H.H. Rosenbrock, "Computer Aided Control System Design", Academic Press, London, 1974.

[5] N. Munro, "The UMIST Control Systems Design and Synthesis Suites", IFAC Symposium on Computer Aided Design of Control Systems, Zurich, Switzerland, 1979.

APPENDIX 1

Nomenclature

F_d	Fuel demand (%)
$G(s)$	Transfer function matrix
$g_{ij}(s)$	ijth element of $G(s)$
I_p	Moment of inertia of propeller plus propeller shaft
I_T	Total moment of inertia of propulsion plant (referred to propeller shaft)
k_{ij}	ijth gain of $G(s)$
M	Ship mass
N_p	Propeller shaft speed
N_t	Turbine speed
$P(s)$	System matrix
Q_D	Torque developed by Turbine
Q_p	Propeller Torque
Q_s	Shaft Torque
Q_t	Turbine Torque
R_s	Ship resistance
s	Laplace operator
T_p	Thrust developed by propeller
V_s	Ship speed
ϕ_a	Achieved pitch
ϕ_d	Demanded pitch
τ	Turbine time constant
τ_4	Pitch actuator time constant

AUTOMATIC CONTROL OF LATERAL SEPARATION DURING UNDERWAY REPLENISHMENT

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ABSTRACT

The United States Navy has decided to build and test a prototype manual, display-aided manual, and automatic control system for underway replenishment at sea. In this paper we will discuss some of the practical aspects of the development of a control algorithm for this process. In particular, we will show that "modern" and "classical" control theories can unite to provide an "optimal" approach to system design. The modern method provides the designer with a rapid means of obtaining control system gains for a wide variety of ship dynamics, and the classical approach permits insights and stability measures not easily available from the modern methods.

A simple technique is presented which allows one to bridge the gap from a complex, multi-input/multi-output control system set in a state variable framework to the more conventional phase/magnitude domain. Using this approach Nyquist plots, Bode diagrams, and the other frequency domain tools become available to the state space formulation. This technique is also useful for multi-input/multi-output systems set in a classical framework but for which the block diagram algebra required to obtain gain and phase margins would be extremely cumbersome.

INTRODUCTION

Underway replenishment (UNREP) operations require a great deal of skill and seamanship to accomplish successfully, especially in heavy seas where the probability of collision is high. The availability of high reliability/high capacity micro-processors and sensing systems, provides a means to aid this complex task. Therefore, the U.S. Navy has decided to investigate a prototype system for manual, display-aided manual, and automatic control during UNREP. The system must be able to handle both lateral and longitudinal control during approach, tracking, and breakaway phases of the manuever.

In this preliminary paper we shall discuss lateral control, and even that only succinctly, and leave the longitudinal (speed) control for later documentation. Design procedure considerations, the controller's general structure, and the methods used to analyze the resultant control system will be presented. It should be noted that the design of a practical control algorithm requires substantially more effort than merely determining a set of gains that will stabilize the resulting closed loop system. User options, error handling, and system protection are all in the province of control system development. As a simple example, consider what must be done when bad data is discovered from a sensor. Only the algorithm designer can address the question of what to do in the presence of bad data; how long can the system function with bad data (or in the absence of data), and what percentage of bad data is permissible. Unfortunately, we cannot study these highly important details at this time but must focus our attention on the more "important" problems of control. However, the system designer must be aware that it is often this ingenuity at handling the "less important" problems that determines the success and acceptability of the final product. With that caveat we will proceed to discuss some of these more important problems.

BACKGROUND

In the United States the majority of work on automatic control during UNREP has been conducted either at the David W. Taylor Naval Ship Research and Development Center (DTNSRDC) or the Navy Postgraduate School. The work of Professor G. Thaler and his colleagues (1,2) has provided several important theoretical results including the fact that the two ships can be considered as essentially independent for control design purposes. Interest and effort at DTNSRDC has been more intense and, over the past several years, has yielded a substantial contribution to the simulation problem (3,4,5,6,7) and some preliminary automatic control system designs (8,9,10). These studies provided the basis for, and demonstrated the feasibility of, automatic control for UNREP. However, several practical problems remained to be solved prior to construction and installation on a service wide basis. These include:

(1) A generic design approach which can be quickly applied across a wide range of ship dynamics.

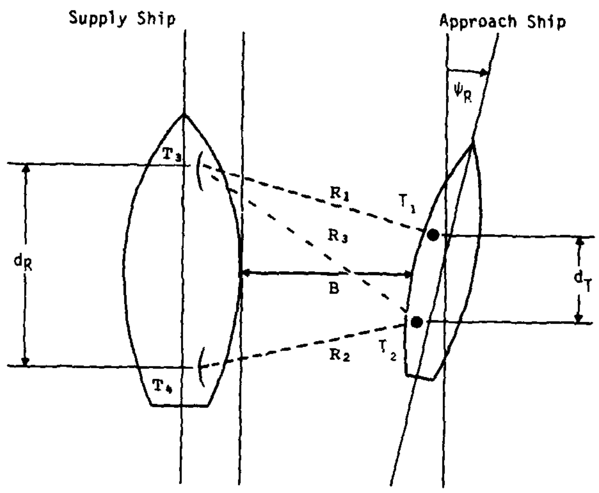
(2) High stability margins to assure insensitivity to uncertainties in the hydrodynamic parameters.

(3) A method to compensate for high frequency effects due to both ships' roll and seaway disturbances.

Each of these factors has had its influence on the UNREP lateral control system design. In the following sections we will describe the control system configuration that was developed with respect to these considerations. In addition, a means for evaluating the stability and robustness of the resulting high order system will be discussed.

HIGH FREQUENCY EFFECTS

A schematic diagram of a sensor system for UNREP is shown in Figure 1. The receiving ship has at least two transmitters which broadcast a high frequency beam (radar and laser are both possibilities) at two corresponding targets on the supply ship. The supply ship attempts to maintain a steady course and speed while the receiving ship is tasked with maintaining station with regard to the supply ship. This procedure is instituted because the receiving ship is usually the smaller of the two and more maneuverable. Once the distances R_1, R_2 and R_3 are measured by the sensor system it is simple to compute the lateral separation and relative heading of the two ships. These two measurements are the basic inputs to the Close Range Ship Control (CRSCS). Both of these inputs are corrupted by high frequency "noise" which must be removed via digital filtering techniques before they can be used for effective control. The sources of this noise are first order wave effects which primarily influence relative heading measurements and the roll characteristics of both the supply and



T_1 = transeiver 1

T_2 = transeiver 2

T_3 = transponder 1

T_4 = transponder 2

d_R = distance between transponders

d_T = distance between transeivers

R_1 = measured distance between T_1 and T_3

R_2 = measured distance between T_2 and T_4

R_3 = measured distance between T_2 and T_3

B = Lateral separation between ships

ψ_R = relative heading of approach ship with respect to supply ship

Figure 1. Schematic Diagram of UNREP Sensor System

and receiving ships which primarily influence lateral separation measurements. First order wave effects cause high frequency yaw motion of both ships for which the control (rudder) cannot compensate. Additionally, roll motions can cause apparent changes in measured lateral separation when no change in the actual lateral separation of the centers of gravity of the ships has occurred.

The seaway and roll disturbances tend to be narrow band random processes. First order seaway forces and moments will occur at frequencies similiar to wave encounter frequencies. Thus, because the sea tends to resemble a narrow band process in many instances, perturbations in heading (and to a lesser extent, lateral separation) occur at sea encounter frequencies. Both ship's roll dynamics can be thought of as lightly damped second order systems. When subjected to any reasonably broad band forcing function, even the relatively narrow band sea forces, the primary responses will be at the natural roll frequencies of the two vessels.

One of the easiest methods to reject narrow band noise from a signal is via the use of a "notch filter". A notch filter is a dynamic system whose response is the neighborhood of some particular frequency is considerably attentuated while response at frequencies away from the notch are unaffected. The simplest notch filter is the second order form:

$$N(s) = \frac{s^2 + W^2}{s^2 + 2ZWs + W^2} \tag{1}$$

Where: s is the Laplace transform variable
W is the notch filter frequency
Z is the damping ratio of the notch denominator

Since the lateral separation loop is primarily corrupted by roll motions and because the roll frequency of the receiving ship is fixed and known, it is a simple matter to select the notch frequency for the lateral separation filter to be the receiving ship's natural roll frequency. Of course, this will leave a residual apparent motion due to the roll of the supply ship. However, the fact that most ship's roll frequencies are quite similar implies that the introduction of the above notch will also attentuate the supply ship's roll effects to some extent.

The first order wave effects occur at frequencies that depend on sea state, ship's heading with respect to the sea, and ship speed. For these reasons it is not possible to select a single notch filter frequency to compensate for all possible variations. We have, therefore, introduced an adaptive mechanism for selecting the relative heading notch frequency. The nature of this mechanism is too complicated to explain in detail here but in essence is a technique which adjusts the notch frequency to minimize the filter output power.

Naturally, there will be considerable input power at frequencies too high for effective control but located in the region for which the notch filters do not have significant effects. Other high frequency noise effects are due to such items as complex multi-directional seas, effects of pitch and heave motions on relative heading and lateral displacement measurements, and other ship's roll (as mentioned above). To attentuate these disturbances additional low pass filtering must be introduced. However, this filtering must be added in such a way that adequate stability margins are maintained to insure robustness of the total control system. This is described in the next section.

LINEAR-QUADRATIC CONTROL

In order to meet the design goal of establishing an overall procedure that is quickly and easily adaptable across a varying range of ship dynamics, we elected to use the Linear-Quadratic-Gaussian (LQG) control approach. LQG theory is a well known part of what has come to be known as "modern" control theory (11,12). Basically, this theory states that if we have a linear system (that is, one described by a set of linear differential equations) which is disturbed by Gaussian white noise, then the optimal control is a set of gains multiplied by the system states when the performance functional to be minimized is a quadratic form. For most practical systems a quadratic form can be thought of as a sum of mean square values of the states.

The concept of the "state" of a system is one of the key notions of modern control theory. The system to be controlled must be expressed as a set of first order linear differential equations and the differentiated variables are referred to as the "system states" or simply the "states". For the UNREP lateral control system, the "states" are lateral range, lateral range rate, relative heading, and relative heading rate.

The basic result of LQG theory can be stated mathematically as follows:

The control which minimizes the performance functional:

$$J = E\left\{x^T Q X + u^T R u\right\} \tag{2}$$

subject to the constraint:

$$\dot{x} = Ax + Bu + Fw \tag{3}$$

is a linear combination of the states:

$$u = -Gx \tag{4}$$

where $G = R^{-1}B^T K$ (5)

and K is the symmetric, positive definite solution of the matrix Riccati equation:

$$-KA - A^T K - Q + KBR^{-1}B^T K = 0 \tag{6}$$

with the following definitions:

E is the expected value operator
T denotes vector or matrix transpose
x is the state vector (e.g., lateral range rate, lateral range, relative heading rate, and relative heading)
A is the system matrix made up of ship's hydrodynamic coefficients
B is a matrix relating the control (i.e., rudder) influence on the ship's states
w is a white noise disturbance
F is a noise scaling matrix

The major power of the LQG approach is that it is extremely flexible and adaptable. The only design decision to be made is the choice of the weighting matrices, R and Q, in the quadratic performance functional. Once this choice is made the entire LQG design can be made for many ships at all speeds with little effort on the part of the designer other than entering ship's parameters into the computer programs. However, the choice of R and Q is not always apparent (although practice improves one's ability in this regard) and the evaluation of the resulting design is not simple. In fact, one of the major criticisms of the state variable approaches is that they do not provide the designer with the insights that are obtained using the classical approaches. In a subsequent section we will describe a means for mitigating that criticism.

A considerable amount has been written with regard to how LQG control theory may be adapted to systems which seem to violate its basic tenets (e.g., non-white noise as in our case). The only serious question here is the extent to which a surface ship's motions can be described by a linear system as required by the theory. However, we have found in our experience that the linear description is certainly adequate for at least initial control system design which is then later evaluated using full non-linear simulation techniques.

As one example of the flexibility of the LQG approach consider the fact that we must also minimize the rate of rudder motions because of inherent rate limits in the hydraulic system. That is, we wish to minimize a performance functional of the form:

$$J = E\left\{x^T Q x + u^T R u + \dot{u}^T N \dot{u}\right\} \tag{7}$$

This is easily done by defining a new state, x_1, as:

$$x_1^T = \left[x,\ u\right] \tag{8}$$

and a new control:

$$u_1 = \dot{u} \tag{9}$$

The new optimal control problem has the solution:

$$u_1 = -G_1 x_1 \tag{10}$$

By partitioning G_1 properly we obtain:

$$\dot{u} = \begin{bmatrix} -G & -G_u \end{bmatrix} \cdot \begin{bmatrix} x \\ u \end{bmatrix} \tag{11}$$

which can be re-arranged (via the Laplace transform) as:

$$u = \left(Is + G_u\right)^{-1} G_x x \tag{12}$$

If u is a scalar, as in the control of a surface ship, the result of introducing weighting on the rudder rate is the introduction of a first order filter whose time constant is the reciprocal of G_u. The benefit of this formulation is that the state gains, G_x, automatically compensate for the control filtering. In this application we determined that additional low pass filtering would be required due to the expected presence of considerable quantities of unpredictable noise as described earlier. It is easy to show that including a weighting on control acceleration has the effect of introducing a second order filter in the compensation.

Finally an integral control of lateral separation is added to compensate for the effect of bias forces caused by ship interaction and line tensioning devices. This completes the lateral separation control law and we can now begin to select system parameters and evaluate system stability. Figure 2 is a block diagram of the resulting control system using the subscript "H" for the relative heading loop and "Y" for the lateral separation loop. In its final form the controller is a standard PID (Proportional-Integral-Derivative) control on lateral separation and PD control on relative heading, both proceeded by their respective notch filters. The total system is of fairly high order consisting of the following items:

Description	Order
Ship's states: lateral range and rate, and relative heading and rate.	4
Rudder dynamics	1
Lateral range notch filter	2
Relative heading notch filter	2
Second order filter introduced by weighting control rate and acceleration	2
Integral control for lateral separation	1
TOTAL	12

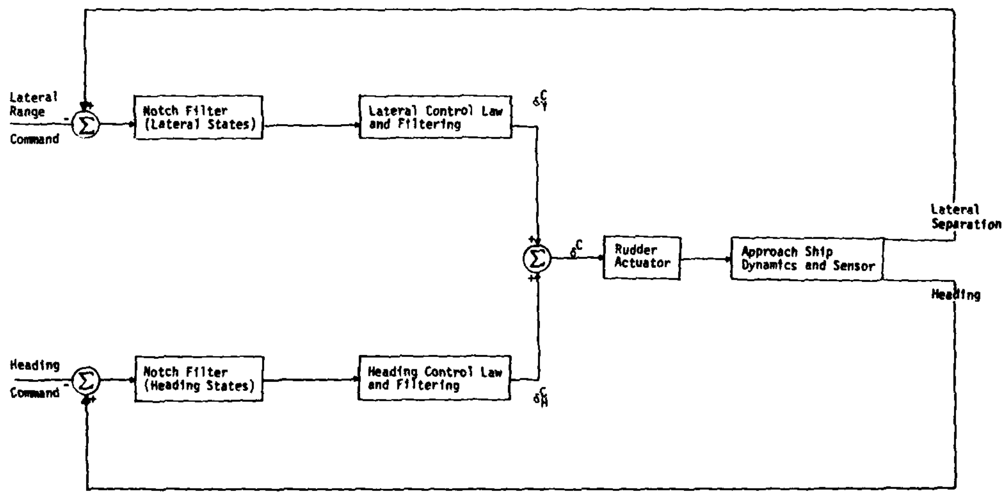


Figure 2. Block Diagram of the UNREP Lateral Control System

STABILITY EVALUATION

In this section we will present a technique for relating multi-input/multi-output, state variable control systems to classical control design methods. This will allow us to take advantage of the insights to be gained from measures of system stability such as phase and gain margins which are not available from the "modern" approaches. Additionally, this technique is simple to understand and implement and does not require extensive manipulation of system blocks and high order Laplace transforms. In fact, in most cases the required matrices can be written from inspection of the system block diagrams.

Because the Linear-Quadratic-Gaussian control approach does not guarantee stability margins (although it does guarantee stability), we must investigate the stability of the resulting system design. When using a state variable approach it is customary to examine the eigenvalues of the closed loop system matrix. For example, with the simple system as described in Equations (3) and (4) we would examine the eigenvalues of the matrix (A -BG). However, consideration of system eigenvalues alone has several drawbacks when one is concerned with system robustness as well as stability. These include:

(1) Eigenvalues give information only with regard to the denominator of the characteristic equation and and do not show the effect of zeroes on system performance.

(2) The eigenvalues do not provide measures of stability margins; that is, how much can the system be perturbed before instability will result.

(3) It is not possible to consider each control loop separately and this may be desireable to take into account known uncertainties with regard to particular dynamics or sensor configuration.

The classical control design approaches used the concepts of phase and gain margin as measures of system stability and robustness. The phase margin is the additional phase lag that could be introduced in the forward loop without instability occurring. The gain margin is the additional gain (usually increase) that could be tolerated before an unstable system configuration occurs. Because these measures have obvious physical interpretations, they provide insights into the system performance that can not readily be obtained from the state variable approach. Gain and phase margins are obtained by considering <u>open loop</u> frequency response plots (e.g., Bode plots) and are easily obtained for single input/single output system; however, they must be redefined for multi-input/multi-output (MIMO) system such as the UNREP control system. In MIMO systems we must define the open loop frequency response plots (and the associated stability margins) as the frequency response that would be obtained if only the feedback loop of interest were broken and all other loops were closed. Clearly the amount of block diagram algebra can become unwieldy even for the 12th order system for UNREP lateral separation. Therefore, we have devised a computational technique which allows us to take advantage of the simplicity and elegance of the state variable approach and still obtain the extremely useful insights to be gained from classical analysis.

feedback system:

$$H_0(jw) = \frac{H_{CL}(jw)}{1 \quad - H_{CL}(jw)} \tag{18}$$

Finally we observe that, in all practical systems, many inputs are related to others by differential or integral operators. For example in the present case we have the variables lateral separation, lateral separation rate, and the integral of lateral separation as related (phase) variables. If we wish to obtain the gain and phase margins for the lateral separation loop, all the inputs related to lateral separation must be included. Therefore the actual closed loop response of a particular state, x_k, to its corresponding input, y_k, is:

$$x_k = (t_{kk} + \sum_i s^{N_i} t_{ik}) \; y_k \tag{19}$$

where the summation is across all i for which a derivative or integral relationship with the variable of interest holds. If we define:

$$\overline{t}_{kk} = (t_{kk} + \sum_i s^{N_i} t_{ik}) \tag{20}$$

then the open loop transfer function of interest can be computed from:

$$\overline{p}_{kk} = \frac{\overline{t}_{kk}}{1 - \overline{t}_{kk}} \tag{21}$$

The magnitude and phase of $\overline{p}_{kk}(jw)$ is computed as:

$$\text{MAG(db)} = 20 \log_{10} (\text{Re}(\overline{p}_{kk}))^2 + (\text{Im}(p_{kk}))^2 \tag{22}$$

$$\text{PHASE} = \text{Tan}^{-1} (\text{Im}(\overline{p}_{kk}))/(\text{Re}(\overline{p}_{kk})) \tag{23}$$

where: Re is the real part of the argument
Im is the imaginary part of the argument.

A computer program has been written that accepts as inputs either the matrix set (A,B,G) or the matrix set (A-BG, BG) in order to form Equation (16). The advantage of this second input form is that it is much easier to formulate those matrices, often by inspection, whenever full state feedback is not used, especially if the matrices are quite sparse. For example, for the UNREP control, the (12x12) matrix BG has only 6 non-zero elements and the (12x12) matrix A-BG has only 35, 13 of which are associated with ship and rudder dynamics.

The open loop Bode plot for the lateral separation loop is shown in Figure 3. This is the response of lateral separation to a sinusoidal input in commanded lateral separation as a function of frequency, with no lateral separation feedback and with the heading loop closed.

The phase margin, -180 degrees minus the phase at the 0 db point, is -64 degrees; and gain margin, the amplitude at the -180 degree phase point is -8 db. (This is only an example and is not representative of the details of the actual design.) This would be considered a quite stable system. As an example of how this information could be used, consider a situation in which sensor data would only be available once every 2 seconds. This would induce a phase lag of only about 4 degrees at the cross-cover frequency (the frequency at which the magnitude equals 0 db) of 0.03 radians per second. Thus one could conclude that sensor delays in this loop will not have a significant influence on system stability.

SUMMARY

The complete designing of an aided, manual, and automatic system for UNREP is a complicated and complex process requiring a thorough knowledge of ship and propulsion plant dynamics, hydrodynamics, control theory, human engineering, and simulation. It would be virtually impossible to report the considerable amount of analysis required to complete the design even if the length of this paper were increased by an order of magnitude. Therefore we have attempted to present only a small portion of the problem, that of stabilizing the lateral separation control loop. Even that must be presented in an incomplete manner which only alludes to some of the problems associated with data rejection and adaptive filter construction. This was done with the prime purpose of focusing the intensity of the presentation on some important results of early efforts.

First, for those more theoretically minded, we have devised a simple scheme for relating the modern state variable approach to the classical presentation of open (or closed) loop magnitude and phase information. This data can then be presented in the format most familiar and usable to the designer as either Bode, Nyquist, or Nichols charts. This is extremely important as the modern, that is optimal, approach typically does not give the designer any "feeling" for the quality of the design and its robustness.

The second point, of a more practical nature, is that a real system design must take into account factors that may often be conveniently ignored in theoretical designs. The specific examples presented in this paper were the effect of roll on lateral separation measurement and the effect of high frequency seaway forces and moments on heading measurement, both of which must be filtered since they occur at frequencies well beyond the overall systems closed loop capability. Because both of these tend to be narrow band inputs, they can effectively be removed by the introduction of notch filters as has been described.

Finally, and perhaps most importantly, the authors do not perceive a conflict between the modern state variable optimal control methods and the classical approaches which use gain and phase information for primary control design decisions. We have shown that the simplicity of the optimal control approach provides a technique for quickly determining overall system structure and a reasonable set of control gains. Further, the flexibility of the modern approach allows the designer to

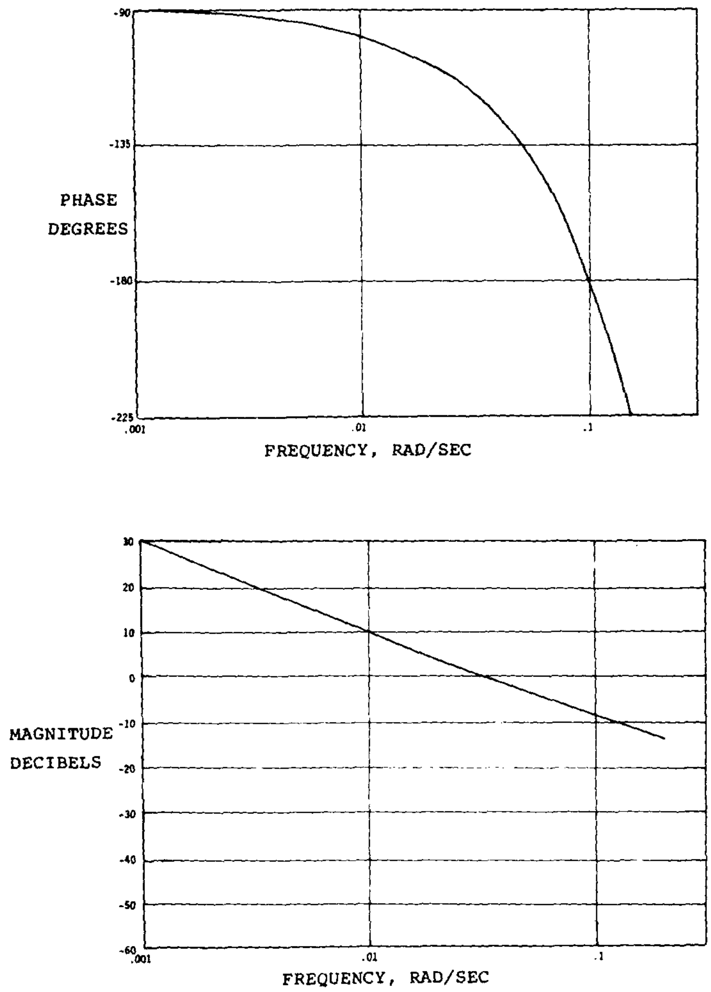


Figure 3. Sample Bode Plot for the Lateral Separation Loop (Open) of the UNREP Lateral Control System.

easily extend the design to include integral control and additional low pass filtering if desired. Despite this flexibility almost any real system design will require the introduction of additional dynamics which are the result of the creativity and experience of the designers. From both practical and theoretical considerations it may not be possible to include these in the optimal control framework. Therefore we must have recourse to the insights that we feel can only be obtained by use of classical techniques. We feel that it is the synergism of the two approaches that is truly "optimal".

REFERENCES

1. Thaler, G.J. and J.J. Uhrin, "Replenishment at Sea: An Automated System", Naval Engineers Journal, December, 1979.

2. Lima, C., G. Astorquiza, and G. Thaler, "Automatic Control for Replenishment at Sea", Fourth Ship Control Systems Symposium, Den Helder, The Netherlands, 1975.

3. Brown, S.H. and R. Alvestad, "Simulation of Maneuvering Control During Underway Replenishment," Journal of Hydronautics, Vol. 12, No. 3, July 1978, pp. 109-117.

4. Dimmick, J.G., R. Alvestad, and S.H. Brown, "Two-Block Romeo: (Simulation of Ship Steering Control for Underway Replenishment)," 28th Vehicular Technology Conference, Denver, Colorado, 22-24 March 1978.

5. Alvestad, R., and S.H. Brown, "Hybrid Computer Simulation of Maneuvering During Underway Replenishment in Calm and Regular Seas," International Shipbuilding Progress, Vol. 22, No. 250, Jan. 1975.

6. Alvestad, R. and S.H. Brown, "Hybrid Computer Simulation of Maneuvering During Underway Replenishment, Phase II," DTNSRDC Report PAS-74-39, Feb. 1975.

7. Brown, S.H., and R. Alvestad, "Hybrid Computer Simulation of Maneuvering During Underway Replenishment," International Shipbuilding Progress, Vol. 21, No. 241, September 1974.

8. Brown, S.H., and R. Alvestad, "Sensitivity Study of Control Parameters During Underway Replenishment Simulations Including Approximate Nonlinear Sea Effects," DTNSRDC Report 77-0003, Jan. 1977.

9. Alvestad, R., "Automatic Control of Underway Replenishment Maneuvers in Random Seas," DTNSRDC Report 76-0048, April 1976.

10. Brown, S.H., and J.G. Dimmick, "Simulation Analysis of Automatic and Quickened Manual Control During Underway Replenishment", DTNSRDC Report 80/007, June 1980.

11. A.E. Bryson, Y. Ho, Applied Optimal Control, 1969, Ginn and Company.

12. H. Kwakernaak, R. Sivan, Linear Optimal Control Systems, 1972, John Wiley and Sons.

COMPARISON OF VARIOUS ADAPTIVE CONTROL TECHNIQUES APPLIED TO AUTOPILOT DESIGN

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ABSTRACT

The increasing need to improve the efficiency and safety of the ship steering process has, in the recent years, stimulated the research and development of adaptive autopilots for surface ships. An adaptive autopilot is designed to optimize both course keeping and course changing under widely varying operational and environmental conditions - with minimum interference from the crew.

To date several different approaches to the design of adaptive autopilots have been proposed. These have resulted from recent developments in the field of automatic control, from new results in statistical identification and modelling of ship motions and from the significant progress in the field of microcomputers.

This paper discusses some of these recently developed techniques and their suitability in relation to the ship steering process.

In addition, the role of simulation in the design of ship control systems is discussed and the importance of adequate mathematical models of ship and environment is underlined.

Finally, results of simulation runs are given, illustrating the main features of one of the adaptive control techniques considered.

1. INTRODUCTION

During the last decade use of automatic control has become familiar on board ships - both in the engine room and on the bridge. Considering the bridge, it is used for automatic station keeping (commonly known as dynamic positioning) of drillships and other offshore vessels and also for automatic track keeping at low speed of pipe laying barges, dredgers and minehunters.

At the same time, the so-called "autopilot" has become an almost standard piece of equipment for automatic course keeping on board merchant and military ships. The classical PID-type autopilot is gradually being replaced by autopilots, which are based on more sophisticated control strategies.

The autopilot with the classical PID-controller meant a considerable improvement in course keeping, provided that the controller was well tuned. The tuning, however, was critical and for that reason it was the weak link of the chain. It often happened that each bridge officer on duty had his own favorite setting of the PID-knobs. In addition, for optimal usage, each speed and loading condition required a separate tuning.

It is not surprising that the idea arose to develop an autopilot, which would require no tuning at all. This implied that the autopilot should be capable of tuning or adapting itself continuously to a changing environment and to varying operational conditions. The fast development of modern control techniques together with the increasing capabilities of digital computers created the condtions under which the first adaptive autopilot could be realized.

At the moment many more of less conventional autopilots are still in use, while on a smaller scale adaptive autopilots are gradually being introduced either on an experimental basis or sometimes even fully operationally.

The first objective of this paper is to emphasize the recent developments in adaptive automatic control, to compare some of the adaptive control strategies used,to indicate aspects which still need to be improved. The wide scope of the paper does not allow too much detail. However the principal characteristics of various adaptive autopilots will be emphasized together with their differences and similarities.

The paper starts with a detailed review of the autopilot design problem, section 2. The general criteria which should be used are discussed in section 3. Section 4 is really the "heart" of the paper: First - as an introduction - attention is paid to the LQG-controller, although not an adaptive strategy. Following this, the adapted PID-autopilot and the model reference and self-tuning concepts are presented.

Simulation and its role in the design process of ship control systems are the subjects of section 5. The development of mathematical models of ship and environment is discussed,while the possible introduction of a so-called Moment Allocation Logic, as a part of the autopilot, is put forward. Some results of simulation runs carried out in the recent years are presented in section 7.

2. THE AUTOPILOT DESIGN PROBLEM

Before we focus on autopilots, we recall a definition of adaptation, as given by Tsypkin (1): "Adaptation is the process of changing the parameters, structure and possibly the controls of a system on the basis of information obtained during the control period, so as to optimize - from one point of view or another - the state of the system, when the operating conditions are either incompletely defined initially or changed". The optimization may be achieved by minimizing a predetermined cost function or by following a reference model.

Let us now examine a general scheme for adaptive autopilots as it might be obtained by combining the main features of some ship control systems which have been proposed in the recent years. It is not the scheme of an existing autopilot, but rather an outline of what might be realized by taking into account more severe performance requirements, on the basis of the recent advance in the automatic control field as well as in computer technology. From a technical point of view the possible solutions are indeed numerous and, besides economic reasons, they mainly depend on the operational characterisitics of the particular ship, on the available measurement systems and on the level of integration of automatic steering with other bridge area functions.

The proposed autopilot, a blockdiagram of which is presented in Figure 1, can be structured from a functional point of view as a three level control system:

At the upper (third) level, we have a decision-making block, of which the main functions are:

- Selection of a proper algorithm corresponding to the steering mode, activated by the operator
- criteria assessment for the identification and control algorithms
- determanition of reference variables
- performance evaluation and monitoring

The steering modes are: course keeping, course changing and track keeping. In this paper we will limit ourselves to the first two modes. Nevertheless, the considerations of this section apply also to the last one. The rudder is considered being the only means of control. Further more, it is assumed that the ships (mean) speed is controlled manually from the bridge.

The criteria assessment for the identification and control algorithms relates to the choice of those a-priori factors which influence the adaptive

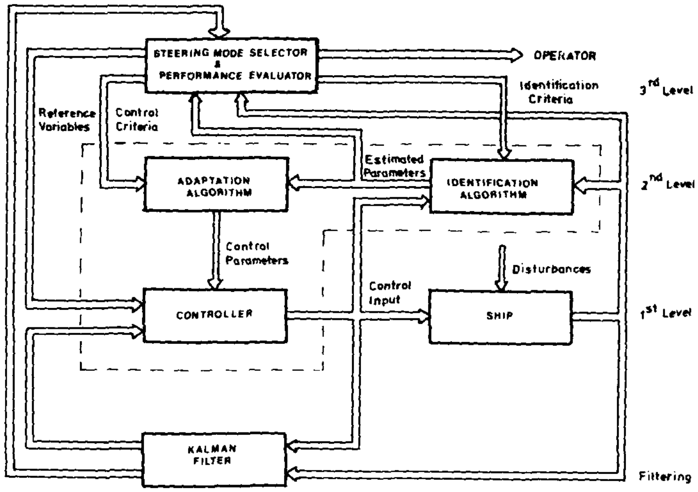

Figure 1 : Blockdiagram of autopilot with hierarchical control structure

autopilot performance. Some of the most important factors concerning the identification algorithms are the initial degree of ignorance with respect to the ship mathematical model (covariance matrix of unknown parameters, non-zero mean values, etc.) the discounting rate of past data during parameter estimation, etc. With regard to the control algorithm, we can mention: weighting coefficients in the feedback control law, saturation limits for the control signal, sampling time interval of the DDC, etc....

The important point is that, once such factors have been assessed, the adaptive autopilot must function unassisted in all applicable operational conditions (speed, loading, trim, etc.) and all environmental situations (wind, waves, current) either stationary or non-stationary.

To evaluate the autopilot performance, a cost function should be defined, which guarantees that the autopilot will function in an optimum or at least suboptimum way in the different steering modes. Such cost functions will be discussed in section 3.

To take into account the parameter variations caused by changing ship dynamics or by non-stationary environments, it is necessary to include also a <u>second</u> functional level, with following tasks:

- Identification of unknown parameters
- Adaptation of the controller parameters with respect to the identified mathematical model

This second level is often based on a "Certainty Equivalence" argument (2), according to which the unknown parameters are estimated on-line by a recursive algorithm and the control law is obtained by replacing the true parameters with their current estimates in the feedback expression. A detailed discussion of this approach will be given in section 4.

It is worth noting that, in general, such adaptive control strategies are not optimal and it is quite complex to examine their main properties analytically, such as convergence characteristics of estimated to true parameters, overall stability and robustness of adaptation and control algorithms. For this purpose it is better to resort to simulation.

Finally, at the first level of the adaptive autopilot blockdiagram there is a

- Control law actuator

which calculates the rudder command signal and feeds it, after a check on saturation conditions, to the ship steering gear.

The adaptive autopilot performance may be improved by means of a multisensor Kalman filtering module, in order to obtain, on the basis of available measurements, the best estimates of the state variables (heading, rate of turn and sway velocity), being the characteristic variables for the ship steering process. Such a filtering module can also be utilized for sensor checking and failure tolerance purposes and it can serve as an aid to integrated navigation systems.

As a final remark, we observe that most of the theoretical results are deduced under the assumption of linear time-invariant systems. It implies that if such control strategies are implemented in an actual design, all effects of descrepancies between the linear, time-invariant process and the real process should be carfully examined.
To check out the design it is once again useful to rely on simulation.

3. GENERAL CRITERIA

With respect to the design of adaptive autopilots, following general criteria are commonly accepted when judging the most important design aspects:

- Performance of the autopilot during course keeping and course changing under varying operational and environmental conditions.
- Capability of the design to deal with the uncertainty in knowledge of the ship steering parameters.
- Operational efficiency (suitability of design for different ships types, manual-auto switching, etc.)
- Simplicity of control algorithms and demands made upon the measuring devices.

The first two criteria will be discussed in this section. The third and fourth criterion will be implicitly dealt with in section 4, where the adaptive control strategies are presented.

Performance criteria.

The performance criterion should be chosen in accordance with the specific task of the autopilot. During course keeping, fuel saving considerations suggest that the propulsion losses should be minimized. As shown by Källström and Norrbin (3), the two main contributions consist of the increase of resistance due to periodic yawing of the ship about the mean course and of the added resistance, caused by rudder motions. This leads to a quadratic cost function (most suitable for optimization procedures!) of the type:

$$J = \frac{1}{T} \int_0^T \left[(\psi(t) - \psi_{ref})^2 + \lambda \delta^2(t) \right] dt \qquad (3.1)$$

where ψ = actual course
ψ_{ref} = reference course
δ = rudder angle
λ = weighting factor

$\psi(t) - \psi_{ref}$ = course error

The factor λ weights the rudder action relative to the course error and depends on the particular ship and on the operational and environmental conditions. Values for λ have been suggested in (3), (4), (5).

During course changing, especially while navigating in congested waters, safety requirements are prevailing and this leads to a cost function which guarantees that the transition from the previous reference course to the new one takes place without overshoots or undershoots. This leads to a quadratic cost function of the type:

$$J_f = \frac{1}{T_f} \int_0^{T_f} \left[\psi(t) - \psi_{ref}(t) \right]^2 dt \qquad (3.2)$$

where $\psi_{ref}(t)$ is a time-varying course set-point which governs the course changing process during the transitions time T_f and depends on the steering characteristics of the ship. It is chosen in such a way that the course changing manoeuvre is executed at a constant rate of turn.

Mathematical models

In the autopilot design process, the choice of an adequate mathematical model of ship and disturbances is very important. It determines the degree of uncertainty, in spite of which the autopilot should be able to perform. The structure of the autopilot may very well depend on the knowlegde of the real process.

The motions of a ship as a rigid body can be described by six coupled non-linear differential equations. (3 translations, 3 rotations). For our purpose only the motions in the horizontal plan (i.e. the longitudinal or surge, lateral or sway- and yawing motion) are observed - see Figure 2.

Roll, pitch and heave are considered to be of secondary importance.

For the course keeping mode of operation a usual assumption is that the longitudinal motion (i.e. the forward speed) of the ship is constant. This assumption allows to linearize the lateral and yawing motion equations at the chosen forward speed. The autopilot design is then based on the linearized model of the real process. Also for the disturbances a simplified model is used in most cases.

Nevertheless, the effect of the forward speed of the ship should be taken into account in the autopilot design. Gain scheduling is a common solution. Section 6 of this paper proposes to include the effect of speed in a so-called Moment Allocation Logic.

The linearized lateral and yawing motions can be described in a standard state space notation, see also Åström (6):

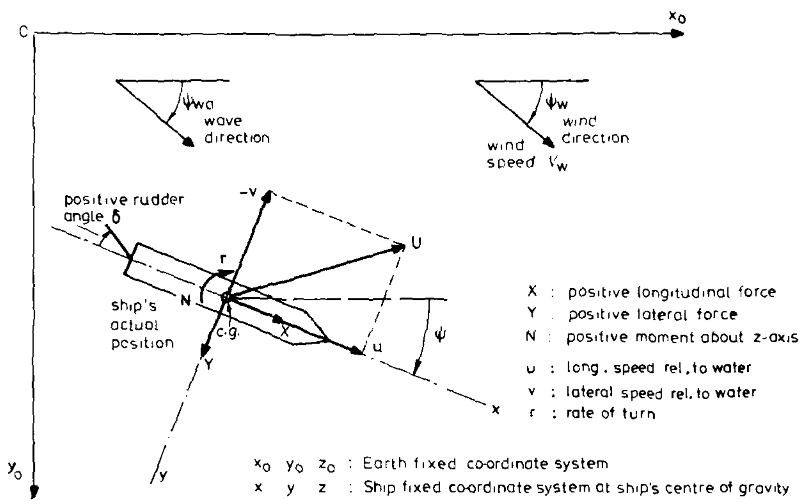

Figure 2 : Co-ordinate systems and sign convention

$$\frac{d}{dt}\begin{bmatrix} v \\ r \\ \psi \end{bmatrix} = \begin{bmatrix} a_{11} & a_{12} & a_{13} \\ a_{21} & a_{22} & a_{23} \\ 0 & 1 & 0 \end{bmatrix} \cdot \begin{bmatrix} v \\ r \\ \psi \end{bmatrix} + \begin{bmatrix} b_1 \\ b_2 \\ 0 \end{bmatrix} \cdot \delta + \begin{bmatrix} d_1 \\ d_2 \\ 0 \end{bmatrix} + \begin{bmatrix} w_1 \\ w_2 \\ 0 \end{bmatrix} \qquad (3.3)$$

Where v, r ψ and δ are the lateral- or sway velocity, the angular velocity, the heading angle and the rudder angle, respectively; the reference course ψ_{ref} is assumed to be zero.

The coefficients a_{ij} and b_i $(i, j = 1,2)$ are functions of the hydrodynamic derivatives and of the ship's forward speed, loading, trim, water-depth, while a_{13} and a_{23} model the disturbances (wind and waves), see Äström (6). The coefficients a_{13} and a_{23} are random variables depending on the angle-of-attack as well as on the spectral characteristics of wind and waves.

We may write: $a_{13} = \dfrac{\partial Y_d / \partial \psi}{m_y}$, in which Y_d is the lateral disturbance force

due to wind and waves, and m_y is the virtual mass of the ship, in lateral direction. If we consider only the contribution of the wind Y_{wi} , a_{13} becomes:

$$a_{13} = \frac{\partial Y_{wi} / \partial \psi}{m_y}$$

Y_{wi} is a function of the lateral wind force coefficient and of the relative wind velocity. Both the coefficient and the relative velocity are functions of the ships heading. In addition, they include the effect of the random wind direction and wind speed fluctuations.

The wind- and wave force and moments at constant heading are given by d_1 and d_2 (constant part, only for small ψ), while $w_1(t)$, and $w_2(t)$ are stochastic processes which model their randomly fluctuating components.

In a general fashion it is possible to assume for them the following representation:

$$\begin{vmatrix} w_1 \\ w_2 \end{vmatrix} = \begin{vmatrix} c_{11} & c_{12} \\ c_{21} & c_{22} \end{vmatrix} \cdot \begin{vmatrix} e_1 \\ e_2 \end{vmatrix} \qquad (3.4)$$

where c_{ij} (i, j = 1,2) are constant coefficients depending on the environments and $e_1(t)$, $e_2(t)$ are independent gaussian white noise processes. A certain simplification is further obtained if the forcing terms $w_1(t)$ and $w_2(t)$ themselves are assumed to be white noise , i.e. if $c_{11} = c_{22} = 1$ and $c_{12} = c_{21} = 0$. Such approximation seems to be acceptable at least for large ships, see Åström (6).

The coefficients a_{ij} and b_{ij} (i, j = 1,2) may be determined from tank-tests or, if possible, from fullscale identification trials. The coefficients a_{13}, a_{23} must be determined on-line, whenever significant environmental changes occur. In view of this, it seems that some improvement could be achieved if direct measurements of sea state and wind were added to the on-board Kalman filtering unit.

From eqs. (3.3) and (3.4) an input-output model may be obtained, which - in the discrete-time domain - assumes the following A.R.M.A. (Auto Regressive Moving Average) form:

$$y(t) + \sum_{i=1}^{n} a_i y(t-i) = \sum_{i=0}^{n} b_i u(t-k-i) + \sigma \left| e(t) + \sum_{i=1}^{n} c_i \, e(t-i) \right| + d \qquad (3.5)$$

where the output $y(t)$ represents the deviation from the reference course and $u(t)$ the commanded rudder angle, while σ is a parameter depending on the external noise level and $e(t)$ is a gaussian zero-mean white noise with known variance. d represents the non-zero mean value of the disturbances. Like for model (3.3) the parameters a_i en b_i, which represent the deterministic part of eq. (3.5), depend on operational conditions, while the σ, c_i and d parameters depend on the environment. Note that any pure time delay in the process is modelled easily by the factor k.

The model order n and the number k of pure time delays are determined by the structure of the state-space model (3.3) and by knowledge of the steering gear dynamics. However, it may often be convenient to regard such a model as a black-box, whose order, delay and parameters are determined by the complex stochastic situation in which the ship steering system is operating. It is possible in such a way to compensate, for example by an increased order n, modelling errors as well as non-linearities, backlash, saturation, etc., which are present in the real ship steering process. This black-box interpretation has been validated by a large number of full-scale identification experiments, (6), (7), (8), (9), (10).

The design of adaptive control strategies based on model (3.5) are generally carried out by STR (Self Tuning Regulator) methods. In section 4 these methods will be discussed in more detail. An important model, which has been used for years as a basis for the design of P.I.D. autopilots is the Nomoto model (11), which is obtained from eq.(3.3) in the absence of the disturbance terms.
We have in that case the transfer function:

$$\frac{\psi}{\delta}(s) = \frac{K(1 + sT_3)}{s(1 + sT_1)(1 + sT_2)} \tag{3.6}$$

which can be reduced in certain situations to the low frequency Nomoto approximation:

$$\frac{\psi}{\delta}(s) = \frac{K}{s(1 + sT)}, \text{ with } T \cong T_1 + T_2 - T_3 \tag{3.7}$$

where the gain K and time constants T and T_i ($i = 1,3$) are again functions of the operational conditions.
A well known extension of the Nomoto-model is the non-linear model, proposed by Bech (12).

Some indications of how P.I.D. autopilots based on the above models may be "adapted" will be given in section 4.

As concerns course changing we observe that, if large manoeuvres are involved, the assumption of linearity is no longer valid. The stochastic models (3.3) and (3.5) can, to a certain extent, handle such non-linearities by assuming that their parameters are time-varying during the course changing process.

4. ADAPTIVE AUTOPILOTS

In this section some recently proposed adaptive control techniques applied to autopilot design will be discussed, namely, adapted P.I.D., MRAS (Model Reference Adaptive Systems) and STR (Self Tuning Regulators).

First of all LQG (Linear Quadratic Gaussian) controllers are presented. LQG-controllers, although not of the adaptive type, constitute a class of optimal solutions within stochastic control theory. Therefore they are often regarded as design reference solutions with which suboptimal adaptive control strategies can be compared.

LQG-controllers

A direct inspection of the linearized models (3.3) to (3.7) and of the cost function (3.1) indicates that the course keeping problem can be solved quite nicely by the LQG stochastic control technique.

In order to carry out a unified presentation, let us consider a discrete-time stochastic model in standard state space representation:

$$\underline{x}(t + 1) = \phi\underline{x}(t) + \Gamma u(t) + \Delta\underline{w}(t) \tag{4.1}$$

$$\underline{y}(t) = C\underline{x}(t) + \underline{e}(t) \tag{4.2}$$

The dimensions of the state vector $\underline{x}(t)$ and observation vector $\underline{y}(t)$ depend on the models which are assumed for the ship dynamics and for the measurement system respectively; $u(t)$ is a scalar control input, i.e. the rudder signal. The disturbance vetors $\underline{w}(t)$ and $\underline{e}(t)$ can be assumed, without loss of generality, as independent zero-mean gaussian white noise processes, having known covariance matrices, while matrices ϕ, Γ, Δ and C have consistent dimensions and are time-invariant.

The cost function, associated with the steady-state course keeping process, is taken to be of the form:

$$J = \lim_{N\to\infty} \frac{1}{N} E\left\{ \sum_{t=1}^{N} \left[\underline{x}^T(t)\, Q\underline{x}(t) + \lambda u^2(t) \right] \right\} \tag{4.3}$$

This cost function is a discrete-time extension of (3.1), which - through a proper choice of the weighting matrix Q - makes it possible to handle more general optimization problems.

If the models (4.1) and (4.2) are perfectly known, it can be shown (13) that an optimal solution exists, which minimizes cost function (4.3) and which is obtained by applying the Separation Theorem: The optimal control strategy is separated in two parts: a state estimator, which produces the best estimate $\hat{\underline{x}}(t)$ of state vector $\underline{x}(t)$, and a linear feedback control law, which gives the control signal as a linear function of the estimated state.

In short, we have for the control law:

$$u(t) = L \cdot \hat{\underline{x}}(t) \tag{4.4}$$

$$L = -(\Gamma^T P \Gamma + \lambda)^{-} \cdot \Gamma^T P \phi \tag{4.5}$$

where P is the unique positive definite solution of the Riccati equation in the stationary case and $\hat{\underline{x}}(t)$ is the best estimate of $\underline{x}(t)$, based on the measurements up to time t, according to the Kalman filter recursive equation:

$$\hat{\underline{x}}(t+1) = \phi\hat{\underline{x}}(t) + \Gamma u(t) + K \cdot \left[\underline{y}(t) - C\hat{\underline{x}}(t) \right] \tag{4.6}$$

where the stationary gain matrix K is obtained in terms of the covariances of vector $\underline{w}(t)$ and $\underline{e}(t)$.

The realisation of a particular autopilot, based on the LQG-technique, is dependent on the chosen mathematical models.

Very recently, Reid and Parent (14) have proposed an autopilot based on model (3.3), which is characterized by the state vector $\underline{x}(t) = [v, r, \psi]^T$ and by a perfect observation mechanism $\underline{y}(t) = \underline{x}(t)$. In their paper a sensitivity analysis of the autopilot with respect to different environmental conditions is carried out for a containership.

A LQG-controller based on the state space representation of the Nomoto model (3.7) has been proposed by Åström (15) . In this case the state vector is of reduced order, i.e. $\underline{x}(t) = [r, \psi]^T$, and only the heading measurement is assumed to be available, i.e. $y(t) = \psi(t) + e(t)$. The resulting optimal control law is shown to be a PD-regulator, applied to a Kalman filtered estimate (4.6) of the partially observable state vector.

Ohtsu and coworkers (16), (17), have investigated an LQG-controller based on a multivariable autoregressive model of the ship dynamics including yawing and rolling motions. The model is a multivariable extension of (3.5) and is obtained through the minimum AIC (Akaike's Information Criterion) identification method (18) and consequently the dimension of the state vector $\underline{x}(t)$ depends on the identified model. The optimal behaviour of such an autopilot is obtained by a suitable choice of the weighting matrix Q coefficients.

A common disadvantage of all the above mentioned LQG-controllers is that, in spite of the attractive form of the feedback control law, they cannot easily be included in adaptive control schemes. For that purpose, an identification of the parameters of model (4.1) should be carried out, whenever the environmental or operation conditions change. Let us observe, moreover, that the updating of the control law parameters may involve some computational burden connected with the numerical solution of the Riccati equation. It is worth noting that the choice of a fast algorithm for determining such a solution may be very improtant, see for example (19).

As shown later on, relevant simplifications can be achieved if suboptimal adaptive control laws are used, which minimize more simple one-step cost functions.

Adapted PID-controllers

The first and until now commonly used autopilots for course keeping are basically three terms-controlllers, including proportional, integral and derivative action, with the corresponding control law:

$$\delta(t) = K_P \left[\psi(t) - \psi_{ref}\right] + K_I \int_0^t \left[\psi(s) - \psi_{ref}\right] ds + K_D \dot{\psi}(t) \qquad (4.7)$$

The PD-part of it is, according to (6), the optimal control law for the deterministic Nomoto-model, given in (3.7), section 3, which minimized a cost function of the type (3.1) for $t \to \infty$. A further restriction is, that the solution is only valid for one particular operational condition in calm weather.

Obviously, measures should be taken if the control law is to be applied under real circumstances during course keeping. We mention:

- additon of low-pass filter or Kalman-filter to cope with the wave-induced yawing motions and with the measurement noise (20)
- adjustment of the K_P and K_D coefficients as functions of the forward speed of the ship (20). This method is called speed-, velocity- or gain-scheduling
- adaptation of the PID-coefficients using the MRAS-technique (26)

An adaptation technique, based on mathematical programming, uses partial derivatives of the cost function (3.1). For this purpose, the cost function is written as follows:

$$J(\underline{\theta}) = \frac{1}{T} \int_0^T \underline{z}^T(t,\underline{\theta}) \cdot R \cdot \underline{z}(t,\underline{\theta}) dt \qquad (4.8)$$

in which $\underline{\theta}$ refers to the control parameter vector $\underline{\theta} = [K_P, K_I, K_D]^T$, and $\underline{z}(t,\underline{\theta})$ is the measured vector $\underline{z}(t,\underline{\theta}) = [\psi(t,\underline{\theta}), \delta(t,\underline{\theta})]^T$ and the weighting matrix $R = \text{diag}[1, \lambda]$. The reference course ψ_{ref} is assumed to be zero.

We refer to the following situation: given heading and rudder observations during the time interval $[0,T]$, in which a PID autopilot has been operating with a fixed control vector $\underline{\theta}_0$, an optimal autopilot setting $\hat{\underline{\theta}}$ is desired to be used in the subsequent time. This new control vector is chosen in such a way that it minimizes $J(\underline{\theta})$.

If we assume that a Taylor expausion of $J(\underline{\theta})$ around $\underline{\theta}_0$, including the second order terms, is accurate enough:

$$J(\underline{\theta}) \simeq J(\underline{\theta}_0) + \frac{\partial J}{\partial \underline{\theta}}(\underline{\theta}_0)(\underline{\theta} - \underline{\theta}_0) + \frac{1}{2}(\underline{\theta} - \underline{\theta}_0)^T \cdot \frac{\partial^2 J}{\partial \underline{\theta}^2}(\underline{\theta}_0)(\underline{\theta} - \underline{\theta}_0) \qquad (4.9)$$

the optimal autopilot setting is given by:

$$\hat{\underline{\theta}} = \underline{\theta}_0 - \left[\frac{\partial^2 J}{\partial \underline{\theta}^2}\right]^{-1}(\underline{\theta}_0) \cdot \frac{\partial J}{\partial \underline{\theta}}(\underline{\theta}_0)^T \qquad (4.10)$$

If a mathematical model of the ship dynamics is available - for example in the Nomoto form - the partial derivatives matrix $\frac{\partial^2 J}{\partial \theta^2}(\underline{\theta}_o)$ and vector $\frac{\partial J}{\partial \theta}(\underline{\theta}_o)$ can be easily computed by taking into account the ship closed loop response.

A discrete time version of the above approach has been suggested by Tosi and Verde (21). Sugimoto (22) has proposed an autopilot which combines PID adaptation with adaptive filtering.

If we have no a-priori knowledge of the ship dynamics the relation between the cost function and the control vector $\underline{\theta}$ cannot be expressed analytically. Solutions for the cost function minimization problem should then be obtained by search methods for finding extrema, see Schilling (23).

Although an adapted PID-controller may show satisfactory performance in the course keeping mode, convergence to the desired optimum is not always guaranteed, due to the disturbances acting on the ship.

Model Reference Adaptive Systems

Among the various types of adaptive control techniques, Model Reference Adaptive Systems (MRAS) are important since they lead to algorithms of relatively little complexity with a high speed of adaptation. On the other hand, a certain degree of a-priori knowledge on the mathematical model of the system to be controlled is always necessary. For a detailed exposition of MRAS approach and of its applications, we suggest the book of Landau (24).

The principle of a MRAS is that the desired behaviour of the adjustable closed loop system is included in a "reference model". The adaptation of the relevant parameters is based on the difference between the actual output of the controlled process and that of the reference model.
The same method can handle as a particular case, the parameter identification problem. The real process is then taken as the reference model.

Before showing, how the MRAS method can be applied to the design of autopilots, a short mathematical introduction is given. For this purpose we consider, in a general way, a state space representation for the real system and for the reference model:

$$\dot{\underline{x}}_p = A_p \underline{x}_p + B_p \underline{u} \tag{4.11}$$

and

$$\dot{\underline{x}}_m = A_m \underline{x}_m + B_m \underline{u} \tag{4.12}$$

where the real system state vector $\underline{x}_p$ and the reference model state vector $\underline{x}_m$ are assumed to have the same dimension, while the same input vector $\underline{u}$ feeds the two systems. Let $\underline{\theta}$ be the vector of unknown parameters of the real system matrices A_p and B_p.

If we define a state error vector $\underline{e}(t)$ given by:

$$\underline{e}(t) = \underline{x}_m(t) - \underline{x}_p(t) \tag{4.13}$$

it is possible to obtain a (non-linear time varying) state error differential equation:

$$\dot{\underline{e}}(t) = A_m \underline{e}(t) + \left[A_m - A_p(\theta)\right] \underline{x}_p(t) + \left[B_m - B_p(\theta)\right] \underline{u}(t) \tag{4.14}$$

The aim of adaptive control is to generate an adjustment law for the unknown parameter vector of the form

$$\dot{\underline{\theta}}(t) = \underline{f}\left[\underline{e}(t), \underline{\theta}(t), t\right] \tag{4.15}$$

based on all available data such that all signals are uniformly bounded and $\lim_{t \to \infty} \underline{e}(t) = 0$.

It can be shown (24) that such an optimal adjustment law can be obtained using Lyapunov functions, with the result:

$$\frac{d}{dt} A_p(\underline{\theta}(t)) = F_A \cdot (P\underline{e}(t)) \cdot \underline{x}_p^T(t) \tag{4.16}$$

$$\frac{d}{dt} B_p(\underline{\theta}(t)) = F_B \cdot (P\underline{e}(t)) \cdot \underline{u}^T(t) \tag{4.17}$$

where F_A and F_B are arbitrary positive definite matrices of consistent dimensions and the matrix P satisfies an equation of the type

$$A_m^T P + PA_m = -Q \tag{4.18}$$

where Q is an arbitrary positive definite matrix.

Van Amerongen and Van Nauta Lemke (25, 26) have developed a MRAS autopilot for both course keeping and course changing. In the course changing mode a second order parallel reference model has been chosen for the closed loop response of the ship. This choice corresponds with the assumption that the ships characteristics can be described by the simple Nomoto-model and that the ship is controlled by a PID-controller.

The state vectors for the process (p) and model (m) are:

$$\underline{x}_p^T(t) = \left[\psi,\ r\right] \text{ and } \underline{x}_m^T(t) = \left[\psi_m,\ r_m\right] \tag{4.19}$$

The control vector for both process and model is:

$$\underline{u}^T(t) = \left[\psi_{ref},\ 1\right] \tag{4.20}$$

where $\psi(t)$ and $\psi_m(t)$ are the measured and reference model course angle, $r(t)$ and $r_m(t)$ are measured and reference model angular velocity, ψ_{ref} is the new desired course.

The corresponding dynamics are described by coefficient matrices:

$$A_p = \begin{bmatrix} 0 & 1 \\ \frac{-K_P K}{T} & -\frac{1+K_D K}{T} \end{bmatrix} \quad ; \quad B_p = \begin{bmatrix} 0 & 0 \\ \frac{-K_P K}{T} & \frac{K_I - K}{T} \end{bmatrix}$$

$$A_m = \begin{bmatrix} 0 & 1 \\ \frac{-K_{pm}}{T_m} & \frac{-1}{T_m} \end{bmatrix} \quad ; \quad B_m = \begin{bmatrix} 0 & 0 \\ \frac{K_{pm}}{T_m} & 0 \end{bmatrix} \tag{4.21}$$

$\underline{\theta}^T(t) = [K_p,\ K_I,\ K_D]$ represents the unknown PID-controller gain vector and K_{pm} and T_m specify the desired course changing manoeuvre.

A simple choice of matrices F_A and F_B in diagonal form give the adaptation law (4.16) and (4.17) for the unknown vector $\underline{\theta}$:

$$\frac{d}{dt}\begin{bmatrix} K_p \\ K_I \\ K_D \end{bmatrix} = \begin{bmatrix} f_1\ (p_{12}e + p_{22}\dot{e}) \cdot (\psi(t) - \psi_{ref}) \\ f_2\ (p_{12}e + p_{22}\dot{e}) \\ f_3\ (p_{12}e + p_{22}\dot{e}) \cdot r(t) \end{bmatrix} \tag{4.22}$$

where the coefficients p_{12}, p_{22} are elements of matrix P of (4.8). The parameters f_1, f_2, f_3 are obtained from matrices F_A and F_B and determine the gains of the adaptation mechanism.

Because of non-linearities in the process (limited rudder angle, rudder speed,etc.) a series reference model has to be added to the parallel reference model. The series model modifies the input signal for both the "adjustable system" and the parallel reference model, to guarantee a linear behaviour.

The complete structure of the MRAS in the course changing mode is presented in Figure 3. The "adjustable system" includes the autopilot, whose coefficients are to be adapted, and the real process (i.e. steering gear and ship).

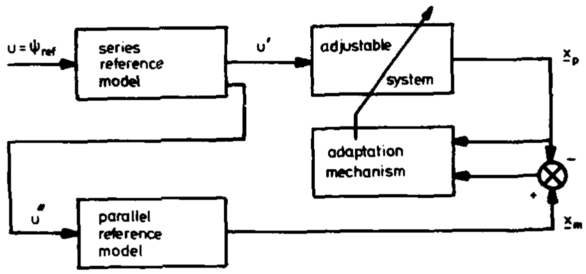

Figure 3 : Simplified block diagram of MRAS in course changing mode

Another significant problem is the noisy measurement of the state variables. Figure 3 suggests that x_p is perfectly known, which is not true. This problem is also present in the course keeping mode.

In the course keeping-mode the MRAS identifies the unknown parameters (K,τ) of the assumed Nomoto-model with the ship as reference model. At the same time the state variables (ψ, $\dot{\psi}$) are estimated. To improve the estimation the measured course and rate-of-turn are filtered. For this purpose, the measured signals are combined with noise free estimates provided by a second adjustable model. The measurements and the estimates are combined and weighted. The weighting factor depends on the noise level of the measured signals. The second adjustable model is required to avoid interference with the identification.
Figure 4 shows a block diagram of the MRAS-structure in the course keeping mode. It is assumed that both the ship's heading and rate-of-turn are measured.

As a direct application of LQG-control, a PD-autopilot is used, based on the identified parameters K and τ.

Summarizing it can be concluded, that the MRAS-autopilot, owing to its fast adaptation, works well during course changing. The suitability for course keeping is less obvious, since the Nomoto on which the reference model is based, does not accomodate the external disturbances and may become too sensitive without the addition of separate filters. Van Amerongen (26) shows that, once such filters have been introduced, the MRAS-autopilot works very well.

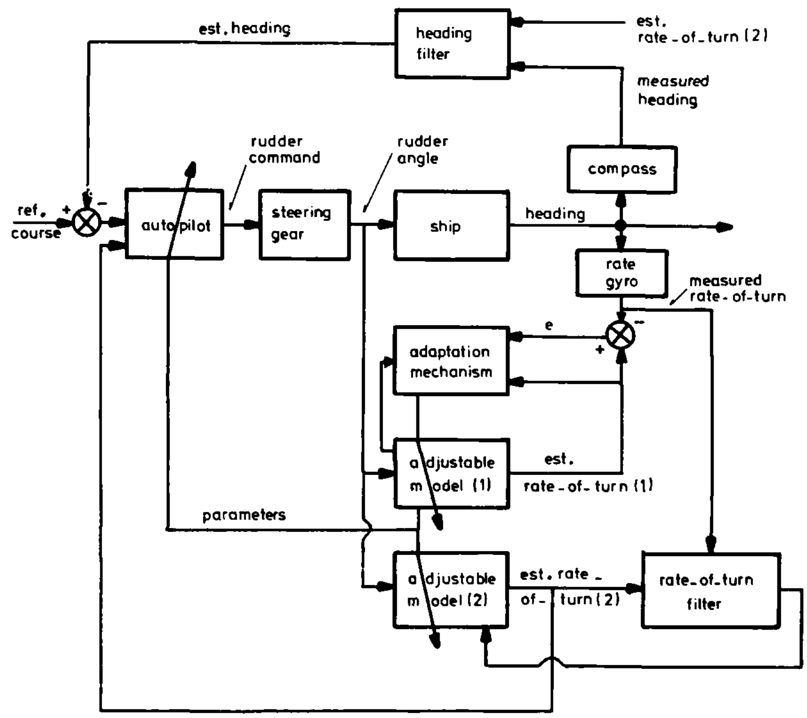

Figure 4 : Blockdiagram of MRAS for course keeping

Self-Tuning Regulators (STR)

Self-Tuning Regulators (STR) are one of the most interesting developments in the adaptive control field of the recent years. Within this class, one finds many different solutions of the problem as the result of different combinations of model structures, identification schemes and controller design, see Åström (27, 28).

All STR-strategies are based on the Certainty Equivalence-principle, which means that the optimal control law for the deterministic problem, with known parameters, is used for the stochastic problem, while the unknown parameters are replaced by their current estimates.

In the original approach the only objective was to minimize the variance of the output of an ARMAtype stochastic system. Eq. (3.5) in section 3 represents such a single input - single output ARMA-system, with unknown but constant parameters.

The method was subsequently extented (29) in order to minimize more general cost functions, including also the systems input and a time dependent set point. In this way the STR-function became two fold and could now also be used as a servo-controller. Eq. (4.23) presents the formula of the extended one-step cost function - see also eq. (3.1):

$$J = E\left\{ [y(t+k) - w(t)]^2 + \lambda u^2(t) \right\} \qquad (4.23)$$

In our case, $y(t)$ represents the deviation from the reference course, while $w(t)$ is the time dependent part of the course set point, which allows us to apply the STR-strategy also to course changing. $u(t)$ is the commanded rudder angle and λ is the weighting factor.

The adaptive control law is obtained by minimizing the variance of the "generalized output variable" $\phi(t)$, which is given by the expression

$$\phi(t) = y(t) + \frac{\lambda}{\beta_0}\, u(t-k) - w(t-k) \qquad (4.24)$$

β_0 is a gain factor and is the estimate of b_0 in eq. (3.5). β_0 should be made a function of the ship's forward speed, according to (20).

The minimization of the variance of $\phi(t)$ is accomplished by an online algorithm, which is constituted by:

- recursive estimation of the parameters of a predictive model obtained by taking into account the closed-loop behaviour of the system
- actuation of the "generalized output" minimum variance control law, which is obtained by setting to zero the k-step-ahead predicted value of the generalized output variable.

It can be easily verified that it is sufficient to predict only the component of $\phi(t)$ due to $y(t)$ which is given by the linear model:

$$\hat{y}(t) = \underline{\theta}^T \cdot \underline{x}(t-k) + \beta_0\, u(t-k) \qquad (4.25)$$

where $\hat{y}(t)$ denotes the prediction of the output variable $y(t)$ at time t, while the unknown but constant parameter vector $\underline{\theta}$ is of the form:

$$\underline{\theta}^T = \left[\alpha_1, \ldots \alpha_n; \beta_1, \ldots \beta_l; h; \delta\right] \quad l = n + k - 1 \qquad (4.26)$$

and $\underline{x}(t)$ is a vector of measured past values of inputs, outputs and setpoints:

$$\underline{x}^T(t) = \left[-y(t), \ldots -y(t-n+1); \beta_0 u(-1), \ldots, \beta_0 u(t-l); w(t); 1\right] \qquad (4.27)$$

The optimal control law, which minimizes the cost function (4.23) is given by

$$u(t) = -\frac{1}{\beta_0 + \lambda/\beta_0}\left[\underline{\theta}^T \cdot \underline{x}(t)\right] \qquad (4.28)$$

The recursive least squares estimate of the parameter vector $\underline{\theta}$ is obtained by a Kalman filter algorithm of the form:

$$\underline{\hat{\theta}}(t) = \underline{\hat{\theta}}(t-1) + \underline{K}(t) \cdot \left[y(t) - \underline{\hat{\theta}}^T(t-1)\cdot\underline{x}(t-k) - \beta_0\, u(t-k)\right] \qquad (4.29)$$

$$\underline{K}(t) = P(t-1)\underline{x}(t-k) \,/\, (\eta + \underline{x}^T(t-k)\cdot P(t-1)\cdot\underline{x}(t-k)) \qquad (4.30)$$

$$P(t) = \frac{1}{\eta}\left[I - P(t-1)\,\frac{\underline{x}(t-k)\,\underline{x}^T(t-k)}{\eta + \underline{x}^T(t-k)\cdot P(t-1)\cdot\underline{x}(t-k)}\right]P(t-1) \qquad (4.31)$$

when $\underline{\hat{\theta}}(t)$ denotes the l.s. parameter estimate at time t.

We note that the estimation algorithm (4.29) to (4.31) can handle also time varying parameters with a constant drift, by means of the exponential forgetting factor η which discounts old data according to the approximate formula:

$$p = \frac{1}{1 - \eta} \quad ; 0 < \eta \leq 1 \tag{4.32}$$

where p is the number of sampled data points which are considered by the estimation algorithm at every time instant.
The forgetting factor η is generally specified by the control designer as a constant at the beginning of the estimation process, dependent of the parameter drift, see (33). In case the drift is an (unknown) function of time, the forgetting factor η should be corrected periodically. The correction should be related to the rate of change of estimated parameters in $\underline{\hat{\theta}}$.

In the recent years, many self-tuning autopilots have been proposed. Källström and Åström (20), (30), have successfully developed a minimum variance STR for course keeping for three different tankers. The autopilot has been tested under simulated conditions as well as during full scale experiments. A self-tuning autopilot for course changing has been simulated by Mort and Linkens (31). The ship under study was a Mariner-class cargo ship.

Two simulation studies on self-tuning control of a supertanker in full-loaded and ballast conditions and of a 2nd generation containership have been carried out by Brink and Tiano (32), (33). The simulation included a large number of course keeping and course changing experiments under stationary as well as non-stationary conditions.

The autopilot described in (33) utilizes the same STR algorithm for the two steering modes. During course keeping the course set-point w(t) in cost function (4.23) reduces to zero, while during course changing the rudder is not weighted, i.e. λ=0, and the transition from the initial course to the final desired one is realized by following a prescribed time-varying course set-point.

A block diagram of this self-tuning autopilot is given in Figure 5.

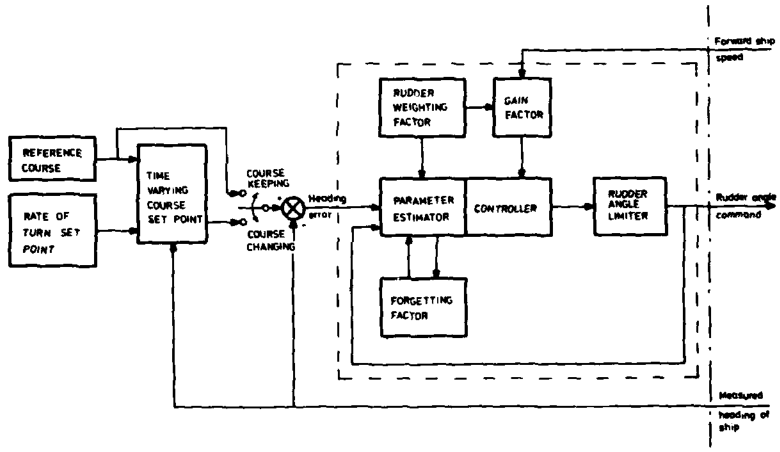

Figure 5 : Simplified blockdiagram of STR - autopilot

Generally the performance of all above mentioned autopilots has proven to be quite satisfactory.

It can be shown, in fact, that the assumed one-step cost function (4.23) is a good approximation of the continuous time cost function (3.1). Moreover these autopilots can take into account the stochastic environment in which the ship operates and, consequently, they can tune parameters in response to the non-stationary characteristics of the disturbances.

The price for the better behaviour of STR-autopilots with respect to other adaptive strategies is a somewhat more complex algorithm, due to a larger number of parameters to be estimated.

The stability and convergence characteristics of the algorithms have proven to be rather good, in comparison with theoretical results.

By means of illustration, we present some results of the simulation reported in (33) also in this paper - see section 6.

5. SIMULATION AS A TOOL IN THE AUTOPILOT DESIGN PROCESS

General aspects of simulation

Simulation is more and more accepted as an - in most cases - indeispensable tool in research and development. The development of fast digital computer has greatly contributed to this growth.

In this section, the role of simulation in design studies is discussed in some detail, considering also those aspects which make that sometimes simulation results do not come up to expectations.

A slightly modified version of the definition, given by Shannon in his book "Systems simulation - the art and science" (34), says: Simulation is the process of designing a model of a properly defined real system and conducting experiments with this model for the purpose of

- understanding the behaviour of the system and/or
- evaluating various strategies for the operation of the system

This definition includes the design or "construction" of the model. Zeigler (35) talks about "modelling and simulation". He distinguishes three elements, see Figure 6, which are related by the two main activities, i.e. modelling and simulation. Two other important activities are validation and verification: modelling should be followed by validation of the model, while making a computer simulation program requires verification of that program with respect to the model.

Although simulation does not inherently need to use a computer, we focus in the context of this paper on simulation, using formal or mathematical models implemented on a computer. Although the so-called simulators are a logical applica-

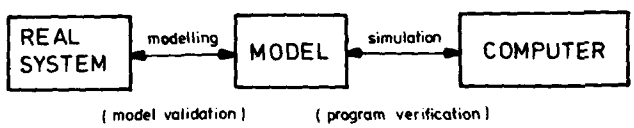

Figure 6 : The basic elements and relations of the simulation process

tion of the simulation technique, they are not discussed. (Simulators are specially of interest for "man-in-the-loop" problems).

The proper definition of the "real system" under study is important: System 1 in Figure 7 can be regarded as an autononous system in a certain environment, while system 2 considers the same system 1 as a subsystem or internal part. The relation between system 2 and 3 can be characterized in the same way.

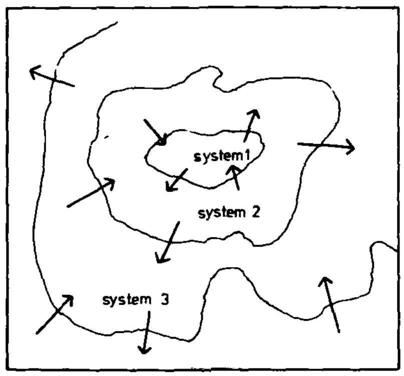

Figure 7 : Systems and subsystems

The systems behaviour is described primarily in the time domain, assuming that we deal with dynamic systems, which functions can be described by some type of differential equations. There are a lot of other characteristics, which may be used to subdivide systems into classes. Some of these are: time-invariant vs. time-varying, linear vs. non-linear, continuous vs. discrete, etc.. Also the process, which is described, can be subdivided by its characteristics, such as: deterministic vs. stochastic, periodic vs. non-periodic, stationary vs. non-stationary.

The mathematical tools differ significantly for the different simulation may very well depend on the assumptions made with respect to the above mentioned characteristics; The designer will, in many cases, disregard certain aspects in order to make his problem more simple, with all possible consequences.

There are a lot more reasons which may lead to an unsuccessful simulation. In a short article in Simulation (36), the authors list "The ten most frequent causes of simulation analyses failure". They also indicate how they should be avoided. Three important conditions which must be fulfilled are:

1) First of all, the simulation objectives should be clearly defined on the basis of a detailed problem definition. The objectives should be realizable! In defining the problem it is necessary to formulate the questions to be answered by the simulation, like:
   - what is to be learned about the system under study
   - what decisions will be based on the results
   - what must be the scope of the simulation in order to satisfy the objectives.
2) In case the (end) user of the simulation results is not the same as who carries out the simulation, the (end) user should be actively participating in all phases of the study.
3) Knowledge and experience should be available in
   - project management
   - modelling
   - computer programming
   - validation of the model: somebody must know how the real system will behave to guide the modelling and judge the validity of the results.

If we consider the design process of an automatic ship control system and the role of simulation in it, we may say that simulation should be an integrated activity of people, who know the physics of the process, the ins and outs of automatic control and of people, who know how to make a flexible computer program and how to set-up and execute an adequate simulation run programme.

The "heart" of the simulation process is the model. It should represent the system in such a way, that the objectives can be achieved. The next section discusses the modelling requirements in more detail.

Modelling requirements

Any model is always a simplified representation of the real system - modelling may therefore be characterized as the art of simplifying. The model should be sufficiently detailed keeping in mind that the model itself is not the goal - it is the means to achieve the goal.

The availability of data, which are necessary to produce quantitative results is of the utmost importance. It may very well determine the degree in which the model can be split up into details: the more details, the more data have to be available.
Also the kind of data is important. If the real system exists, full scale data are very welcome to validate the model. However, in the system design phase, we have to rely on data from literature or obtained from tests with scale models. This makes that simulation results should always be interpreted with caution, because there will always remain some degree of uncertainty.

The ship model. The simulation of a ship controlled by an autopilot distinguishes four components, as illustrated in Figure 8: The autopilot itself, the measuring devices, the environment, acting as disturbance on the ship, and the ship. Small errors in the ship model will lead to bounded errors in the response of the ship, due to the control system. On the other hand: insufficient knowlegde of the process or a too simple model may easily lead to a control system which underestimates the problem or which performance envelope is too narrow.

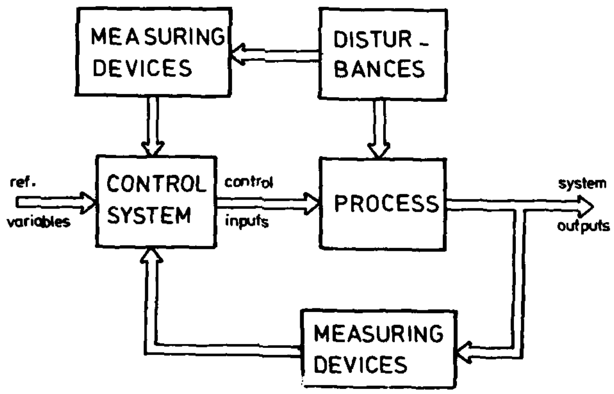

Figure 8 : Basic block diagram of simulation model

In many applications the model is set up too simple. The control engineer, for example, likes to look upon the ship as a linear system. A ship, however, is an inherently non-linear process. Linearization requires to assume a constant speed and allows only small deviations from the chosen equilibrium conditions. A further simplification which is used often is to consider only a deterministic ship model.

For some classes of problems the linearized model approach may yield useful results, although an indefinite discrepancy will remain between the simulated system and the real system. Upon the development of modern control strategies, such as stochastic adaptive control, the process should be modelled even more precisely, especially with respect to the stochastic aspects. The ship's parameters will always - to some extent - be uncertain and they will be time-varying!

It is emphasized that one should think of two ship models. One model will be used for the actual development of the control algorithms (e.g the Nomoto-model in the MRAS and the ARMA-model in the STR) and will probably be a simplified version to be used in the control system itself. The second model should be as realistic as possible, including all non-linear aspects. It replaces the real system and must be capable to detect any weaknesses in the control system design.

The model of the disturbances. Also the model of the environment, which acts as disturbance on the ship, needs careful examination. Wind and waves are stochastic phenomena, characterized by their respective energy - density functions and amplitude probability - density functions. The environment may be stationary or non-stationary. An example of a simulated non-stationary wind speed record is presented in Figure 9.

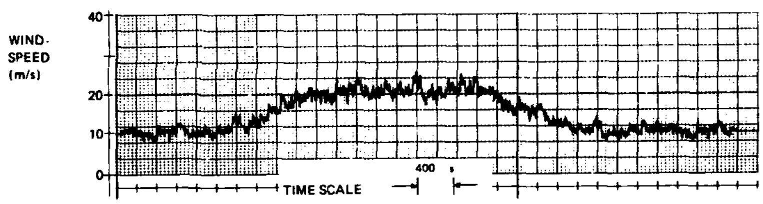

Figure 9 : Non - stationary wind gust signal

The simulation of the wave effects is much more difficult: one problem area is the calculation of the basic linear ship response in regular waves (amplitude and phase versus wave frequency) and subsequently the response in an irregular wave train as a function of time. The second problem area is the simulation of those time responses at varying ship speeds and angles-of-attack to the waves. In other words: It should not only be possible to calculate the time response for a chosen speed and wave angle-of-attack; it should also be possible to change the ship's heading and to increase or decrease speed. At the same time the wave spectrum should be adjustable to simulate the non-stationary character of the sea.

The model of the measuring devices. In most cases it is sufficient, for a proper model of the measuring devices, to implement an error model, characterized by the appropriate energy - density and amplitude probability - density functions. In many cases gaussian white noise is considered an adequate approximation of the real situation. The measurement noise, together with the process noise, should be taken into account in the design of the control system.

The model of the control system. In the design phase, the control system is also represented by a model, although parts of the model may be identical to the real system, especially the software parts. In the design phase, however, the control software is usually written in a high-level programming language and more or less restricted to the basic control algorithms. Upon realisation, the software is likely to be translated, while the operational software is added. During that phase, simulation is as important as always for debugging, for system integration tests and for establishing the performance envelope.

Mathematical model of ship and environment.

The equations of motion of a ship are six coupled non-linear differential equations, with time-and frequency dependent coefficients. The equations of motions, based on Newton's second law, are

$$\underline{F} = m\,(\dot{\underline{U}} + \underline{\omega} \times \underline{U}),\quad \underline{M} = I\dot{\underline{\omega}} + \underline{\omega} \times \underline{I} \qquad (5.1)$$

and describe the ship motions in a ship-fixed co-ordinate system (x, y, z) with origin in the ship's centre-of-gravity, relative to an earth-fixed co-ordinate system (x_0, y_0, z_0), see Figure 2.

$\underline{F}$, $\underline{M}$ are the net force and net moment vector, respectively, acting on the ship; $\underline{U}$, $\underline{\omega}$ are the translational and angular velocity vector, respectively, of the ship relative to the water. m, I are the ship's mass and the matrix of moments and products of inertia, respectibely. The ship's mass and inertia terms are assumed constant, which means that the equations of motions refer to one particular loading condition and centre of gravity location.

The low frequency manoeuvring characteristics in the frequency range between 0-.3 rad/s (roughly) are usually described by a set of non-linear equations with constant coefficients. For our purpose only the motions in the horizontal plane (surge, sway and yaw) are relevant. The motions in waves (frequencies above .3 rad/s) can be described by linear equations, although the coefficients are frequency dependent.

Low frequency manoeuvring equations. The surge, sway and yaw equations of motions are - in general terms - according to Newton's second law:

$$\begin{aligned}
m(\dot{u} - rv) &= \Sigma X = X_{\dot{u}}\,\dot{u} + X_{vr}\,vr + X(U, \beta, r) + X_{ext} \\
m(\dot{v} + ru) &= \Sigma Y = Y_{\dot{v}}\,\dot{v} + Y_{\dot{r}}\,\dot{r} + Y(U, \beta, r) + Y_{ext} \qquad (5.2) \\
I_{zz}\dot{r} &= \Sigma N = N_{\dot{r}}\,\dot{r} + N_{\dot{v}}\,\dot{v} + N(U, \beta, r) + N_{ext}
\end{aligned}$$

The right side of the equations include the hydrodynamic terms, such as the added mass ($X_{\dot{u}}$, $Y_{\dot{v}}$), added moment of inertia ($N_{\dot{r}}$) and cross-coupling terms (X_{vr}, $Y_{\dot{r}}$, $N_{\dot{v}}$) the resistance or (viscous) drag terms (X, Y, N as non-linear functions of U, β, r, or u, v, r) and the external foces and moments (X_{ext}, Y_{ext}, N_{ext}). For example:

$$X_{ext} = X_{rud} + X_{prop} + X_{wind} + X_{waves} + X^1 \qquad (5.3)$$

The components are due to the rudder (X_{rud}), due to propellor(s) or thruster(s) X_{prop}), the environment (X_{wind}, X_{waves}) and any other possible external force (X^1).

The non-linear contributions X, Y, N (U, β, r) combine the effects of the translational velocity U and angular velocity r. There are different ways to split them up into components.

It is obvious that the well known low frequency Nomoto-equation represents only a fraction of (5.2). The X- and Y-equation are neglected (i.e. U is assumed constant and $v = 0$), and only the rudder moment is taken into account. (5.4) shows the linearized moment equation:

$$I_{zz}\dot{r} = N_{\dot{r}}\dot{r} + N(U, o, r) + N_{rud} = N_{\dot{r}}\dot{r} + N_r r + N_\delta \delta \tag{5.4}$$

The transfer function $\frac{r}{\delta}$ (s) follows from (5.4):

$$\frac{r}{\delta}\ (s) = \frac{K}{1+T_s} = \frac{-N_\delta/N_r}{1 - ((I_{zz}-N_{\dot{r}})/N_r)s} \tag{5.5}$$

The gain factor K is the ratio between the rudder effectivity and the damping in yaw, while the time constant is proportional to the virtual mass of the ship.

The extended Nomoto-model observes both the (linearized) Y and N-equations: (5.2) then becomes

$$\begin{bmatrix} (m-Y_{\dot{v}}) & -Y_{\dot{r}} \\ N_{\dot{v}} & I_{zz}-N_{\dot{r}} \end{bmatrix} \begin{bmatrix} \dot{v} \\ \dot{r} \end{bmatrix} = \begin{bmatrix} Y_v & Y_r - mU \\ N_v & N_r \end{bmatrix} \begin{bmatrix} v \\ r \end{bmatrix} + \begin{bmatrix} Y_\delta \\ N_\delta \end{bmatrix} \delta \tag{5.6}$$

Elimination of $\dot{v}$ and v yields the tranfer function

$$\frac{r}{\delta}\ (s) = \frac{K(1 + T_3 s)}{(1+T_1 s)(1+T_2 s)}$$

The gain factor K now includes also lateral motion terms:

$$K = -\frac{N_\delta - N_v Y_\delta / Y_v}{N_r - N_v \left[Y_r - mU\right] / Y_v}$$, and in the same way

$$T_3 = -\frac{(m - Y_{\dot{v}})\ N_\delta - N_{\dot{v}} Y_\delta}{Y_v N_\delta - N_v Y_\delta} \cong -\frac{m - Y_{\dot{v}}}{Y_v - N_v Y_\delta / N_\delta} \tag{5.8}$$

T_1 and T_2 cannot be written explicitly, unless some radical simplifications are made:

$$T_1 T_2 = -\frac{(m - Y_{\dot{v}})\ (I_{zz} - N_{\dot{r}}) - N_{\dot{v}} Y_{\dot{r}}}{N_v(Y_r - mU) - Y_v N_r} \cong \frac{(m - Y_{\dot{v}})\ (I_{zz} - N_{\dot{r}})}{Y_v N_r}$$

and

$$T_1 + T_2 = \frac{(m - Y_{\dot{v}})N_r + (I_{zz} - N_{\dot{r}})Y_v + N_{\dot{v}}(Y_r - mU) + Y_{\dot{r}} N_v}{N_v(Y_r - mU) - Y_v N_r} \tag{5.9}$$

$$\cong \frac{m - Y_{\dot{v}}}{-Y_v} + \frac{I_{zz} - N_{\dot{r}}}{-N_r}$$

The approximations yield the time constants:

$$T_1 \simeq -\frac{I_{zz} - N_{\dot{r}}}{N_r} \quad \text{and} \quad T_2 \simeq -\frac{m - Y_{\dot{v}}}{Y_v}$$, where T_1 corresponds with the

time constant T from (5.5) and T_2 represents the influence of the ship's lateral motion. Eq. (5.8) indicates that T_2 and T_3 are almost equal, which means that, in fact, the lateral and yawing motion have been decoupled again. It also means that the simplification cannot be allowed, in case quantitative results are to be obtained.

Bech (37) has modified the linearized yawing motion by introducing a non-linear term. In this way, also the yawing motions of directionally unstable ships can be studied.

In spite of the advantages of using linear models, it is the authors opinion that the assumption of the constant ship's speed is a constraint, which limits the use of the model too much. The model should not only be used for the initial design of the control system, but also to establish the performance envelope. In that phase, the ship speed will certainly be varied to cover the whole range of operational conditions. Also the environment of wind, waves and current will be varied. Due to the environment the ship speed will change too. And last but not least: When course changing manoeuvres are excented, the ship's speed will drop during the turn. In (32) non-linear models of two large merchant ships have been presented, which may serve as examples.

6. MOMENT ALLOCATION LOGIC

In this section it is proposed to use the knowledge about the hydrodynamic characteristics of the rudder in the controlsystem by introducing a so-called Moment Allocation Logic: This logic should translate a moment command into a corresponding rudder angle command, in the same way as the Thrust Allocation Logic distributes the force and moment commands among the thrusters of a dynamically positioned vessel (38).

The existing autopilots do not calculate the required moment explicitly. They calculate the rudder angle command right away. The drawback is that the influence of the ship's speed must be taken into account by the autopilots overall gain factor.

With the introduction of a Moment Allocation Logic (MAL) it is possible to decouple the calculation of the required moment from the rudder characteristics: At a given moment command, generated by the controller part of the autopilot, the MAL calculates the rudder angle, which produces the required moment (as accurately as possible). The calculated rudder angle command is then transmitted - as usual - to the steering gear.

Figure 10 shows the autopilot set-up with the MAL. The additional advantage of this set-up is that, at the summation-point, between the controller and the MAL, any other known moment acting on the ship may be added in a feedforward loop. These moments may be due to wind, velocity relative to the water, or due to bowthrusters, while it is assumed in this respect that the necessary data are available or can be measured. The input to the MAL is the <u>net</u> moment to be exerted on the ship in order to stay on course or to change course in a prescribed way.

<u>Rudder angle command</u>

To make the function of the MAL more clear a simple example is given, using the linear Nomoto-equation for the yawing motion; without disturbances:

$$(I_{zz} - N_{\dot{r}})\,\dot{r} - N_r r = N_\delta \delta = N_{rud} \qquad (6.1)$$

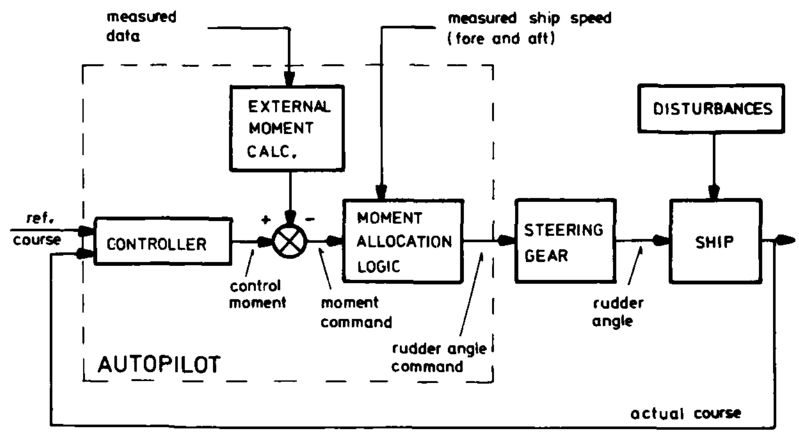

Figure 10 : Basic autopilot set-up with moment allocation logic

N_{rud} represents the moment, due to a rudder angle δ generated by the steering gear in response to the rudder command δ_C. δ_C corresponds with the moment command N_C, according to:

$$\delta_c = \frac{N_c}{N_\delta} = \frac{N_c}{\partial N/\partial \delta} \qquad (6.2)$$

N_C is calculated as a function of the course error, its derivative and/or integral value, etc.. The rudder angle command δ_C depends on N_C and on the rudder characteristics (i.e. the rudder force as function of the rudder angle-of-attack).

The MAL should be able to calculate δ_C. The method to be followed is illustrated below. First the general rudder equations are derived (see also Figure 11)

Rudder moment: $N_{rud} = -\, l * Y_{rud} = -\, l\left[L_R \cos \delta_v - D_R \sin \delta_v\right]$ (6.3)

Rudder lift force: $L_R = C_L(\delta_e)\ \frac{\rho}{2} S_R\, U^2{}_{sr}$ (6.4)

Rudder drag force: $D_R = C_D(\delta_e)\ \frac{\rho}{2} S_R\, U^2{}_{sr}$ (6.5)

Figure 11 shows that $\delta = \delta_e + \delta_v,\ \delta_v = \tan^{-1} \frac{v - lr}{u_{sr}}$

l is the distance between the rudder and the ship's centre-of-gravity.

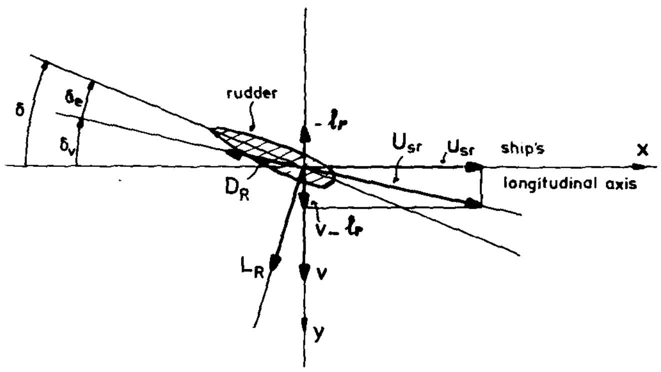

Figure 11 : Rudder - speeds, angles and forces

C_L = rudder lift coefficient, function of δ_e (rudder angle-of-attack)

C_D = rudder drag coefficient, function of δ_e

ρ = density of (sea-)water

S_R = rudder area

U_{sr} = speed of rudder, relative to the water

u_{sr} = longitudinal component of U_{sr}

v_{sr} = v - lr = lateral component of U_{sr}

δ = geometric rudder angle

δ_v = angle between U_{sr} and positive x - axis

δ_e = effective rudder angle or rudder angle-of-attack

Now we can write:

$$N_{rud} = - 1 \frac{\rho}{2} S_R \left[C_L(\delta_e) \cos \delta_v - C_D(\delta_e) \sin \delta_v \right] U^2_{sr} \qquad (6.6)$$

$$= - 1 \frac{\rho}{2} S_R \left[C_L(\delta_e) u_{sr} - C_D(\delta_e) (v - lr) \right] U_{sr} \qquad (6.7)$$

By putting $N_{rud} = N_C$, the effective rudder angle δ_e can be calculated form (6.7), provided that all relevant variables are known. The (geometric) rudder angle command δ_C follows from $\delta_C = \delta_e + \delta_V = \delta_e + \tan^{-1}\frac{v - lr}{u_{sr}}$

At the same time the geometric maximum rudder angle should be taken into account!

First of all the ship's longitudinal and lateral speed relative to the water should be known and secondly the rate-of-turn r. Because the longitudinal speed of the rudder relative to the water, u_{sr}, depends on the slipstream of the propeller, we need also information about the developed thrust and wake fraction. Last but not least, the geometric data and hydrodynamic characteristics of the rudder should be available.

Determination of δ_e and δ_c is done in an iterative way. It may very well be that some of the required data will not be available under practical circumstances, resulting in an approximative calculation. If, for example, only the ship's speed, U, is measured, we have the following expression:

$$N_{rud} = - 1 \frac{\rho}{2} S_R \left[C_L(\delta_e)\right] U^2 \tag{6.8}$$

The rudder angle command δ_e is found by putting

$$N_{rud} = N_c \rightarrow C_L(\delta_e) \rightarrow \delta_e \rightarrow \delta = \delta_c$$

For angles $\delta_e < 20^o$, the lift coefficient C_L can usually be written as a linear function of δ_e : $C_L = C_{L_\delta} * \delta_e$

This makes the relation between δ_c and N_c even more simple:

$$N_{rud} = N_c = - 1 \frac{\rho}{2} S_R \left[C_{L_\delta} \cdot \delta_e\right] U^2 \tag{6.9}$$

and

$$\delta_c = \delta_e = N_c \;/\; - 1 \frac{\rho}{2} S_R \, C_{L\delta} \cdot U^2 = N_c \;/\; C_{N\delta} \cdot U^2 \tag{6.10}$$

(6.10) illustrates that - at a constant moment command N_c - the rudder angle command δ_c is automatically adjusted, when the ship speed changes. A further check should make sure that δ_c stays within the allowable range of rudder angles.

If a Kalman-filter is used to estimate v and r the filter outputs might be used in the MAL.

Moment command. The control algorithms should be updated when introducing the MAL. The conventional PID-algorithm, (4.7), specifies the rudder angle command δ_c, which can be written as follows:

$$\delta_c = K_p \left[\varepsilon_\psi + \frac{1}{\tau_i} \int_0^t \varepsilon_\psi \, dt + \tau_d \dot{\psi} \right] \tag{6.11}$$

The moment command N_c can be written as follows:

$$N_c = K_p{}^1 \left[\varepsilon_\psi + \frac{1}{\tau_i} \int_0^t \varepsilon_\psi \, dt + \tau_d \dot{\psi} \right] \tag{6.12}$$

Using (6.10), the relation between K_p and $K_p{}^1$ is:

$$K_p = K_p{}^1 \;/\; C_{N_\delta} \, U^2 \tag{6.13}$$

(6.13) shows that, without MAL, K_p should be made a function of U^{-2}. With MAL, $K_p{}^1$ can be kept constant, as far as the effect of the rudder is concerned. Quite another problem is the variation of the ship parameters as function of loading, trim and speed. Identification of the parameters and adaptation of $K_p{}^1$, τ_i, τ_d will still be necessary to cope with those variations.

The MAL has not yet been used in the actual design of an autopilot. It is planned to test the MAL first in a simulation design study of a STR-class autopilot. These tests will take place in the near future

7. SIMULATION EXAMPLES

To give some idea of the performance of an adaptive autopilot, a few simulation examples are presented in this section. The examples are part of the work reported in (32) and (33): They show the relevant time - histories of the behaviour of two merchant ships under control of a STR-class autopilot. The two ships are a 220 m container-ship and a 315 m supertanker in two loading conditions.

Figures 12 and 13 illustrate the adaptive course keeping in non-stationary conditions of the containership and of the supertanker - in ballast. Figures 14 and 15 illustrate a number of course changing manoeuvres of the containership and of the supertanker - fully laden.

The weather conditions are summarized in table 1. The non-stationary character of wind and waves was simulated by an increase of both mean wind speed and significant wave height during some time, followed by a decrease down again to the original speed and height. Also the effect of speed-variations was simulated by varying the set rpm, but only during course keeping. During course changing, the weather and nominal rpm-setting were kept constant.

Table 1 Run conditions

task	run number	wind: mean speed (m/sec)	wind: mean direction (deg.)	waves: sign. wave height (m)	waves: wave direction (deg.)	ship: set RPM (%)	ship: ref. course (deg.)
	RTB009	10→20→10	90	4→6→4	90	100	0
	RAT010	10→20→10	150	4→6→4	150	100	0
course keeping	RAT011	10	30	4	30	100→70→100	0
in non-stationary							
conditions	RTB013	10	150	4	150	100→70→100	0
	RTF, RAT017	0	0	0	0	100	-30→30
	RTF, RAT018	10	30	4	30	100	-30→30
course changing	RAT019	10	90	4	90	100	-30→30
in stationary	RTF, RAT020	10	150	4	150	100	-30→30
conditions	RTF, RAT021	20	30	6	30	100	-30→30

(The direction of wind and waves is relative to the ship's longitudinal axis: 0 degrees - wind, waves on the stern; 180 degrees - wind, waves on the bow)

Course keeping

From the time histories of the input-output variables and of two estimated parameters, it can be noted that the adaptive autopilot works very well during all combinations of stationary and non-stationary weather conditions, as well as when the ship's speed is changed considerably.

Course changing

The course changing manoeuvres are characterized by the rate of turn, which is preset by the operator. The preset rate of turn is different for the three ships and all the runs were characterized by an initial parameter estimate equal

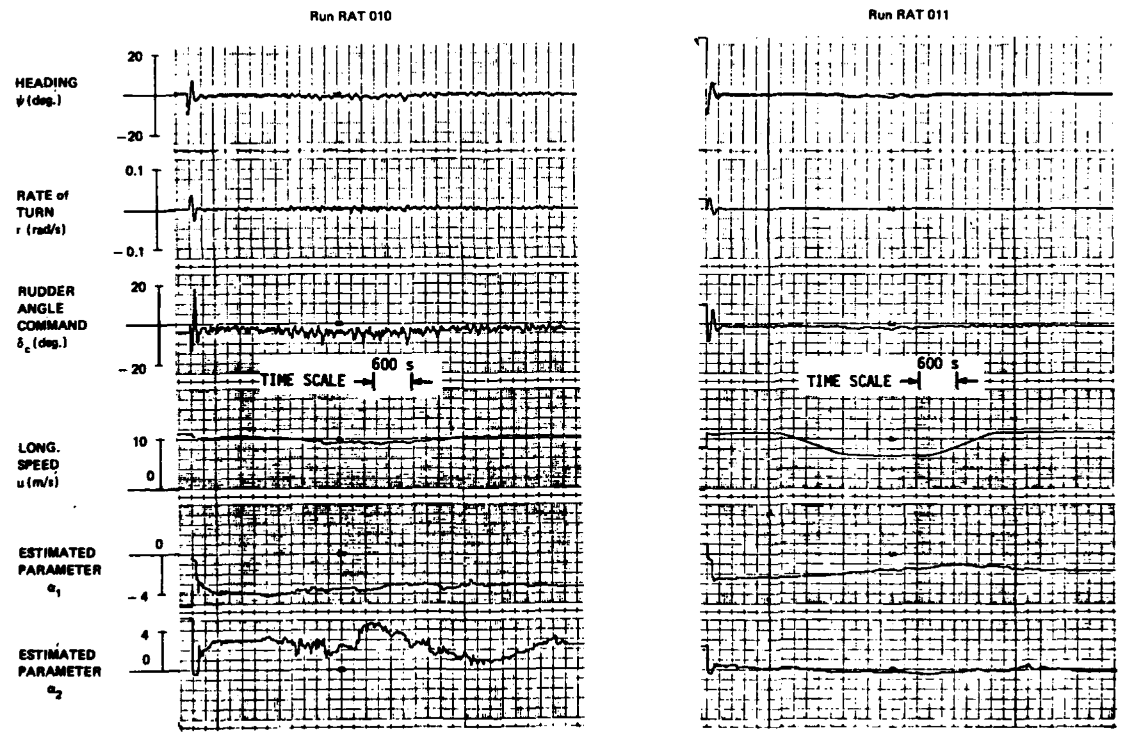

Figure 12 : Adaptive course keeping of the containership in non-stationary conditions (see also Table 3).

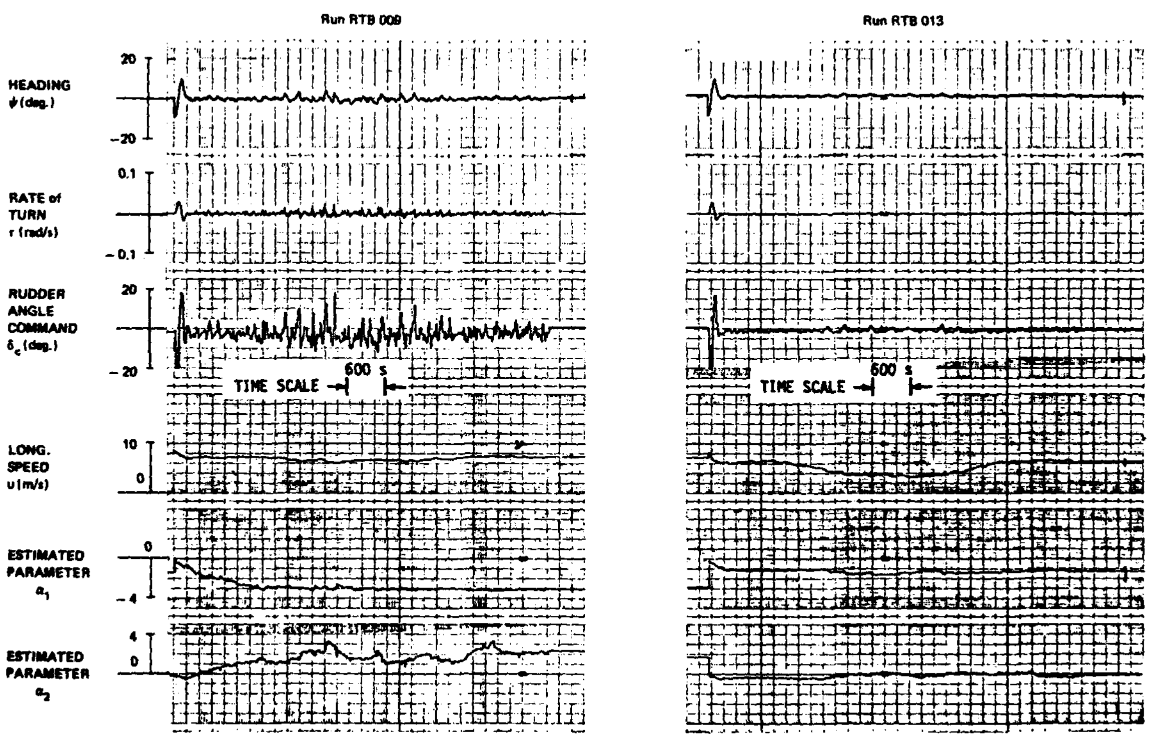

Figure 13 : Adaptive course keeping of the supertanker (in ballast) in non-stationary conditions.

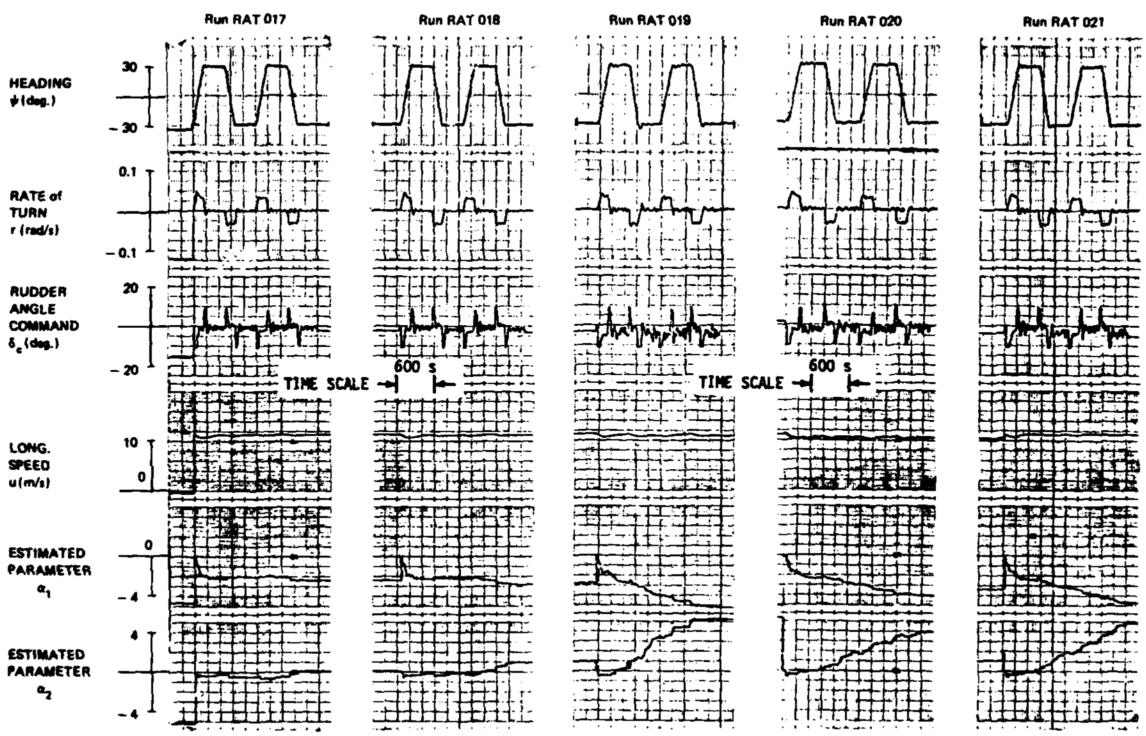

Figure 14 : Adaptive course changing of the containership in different stationary conditions

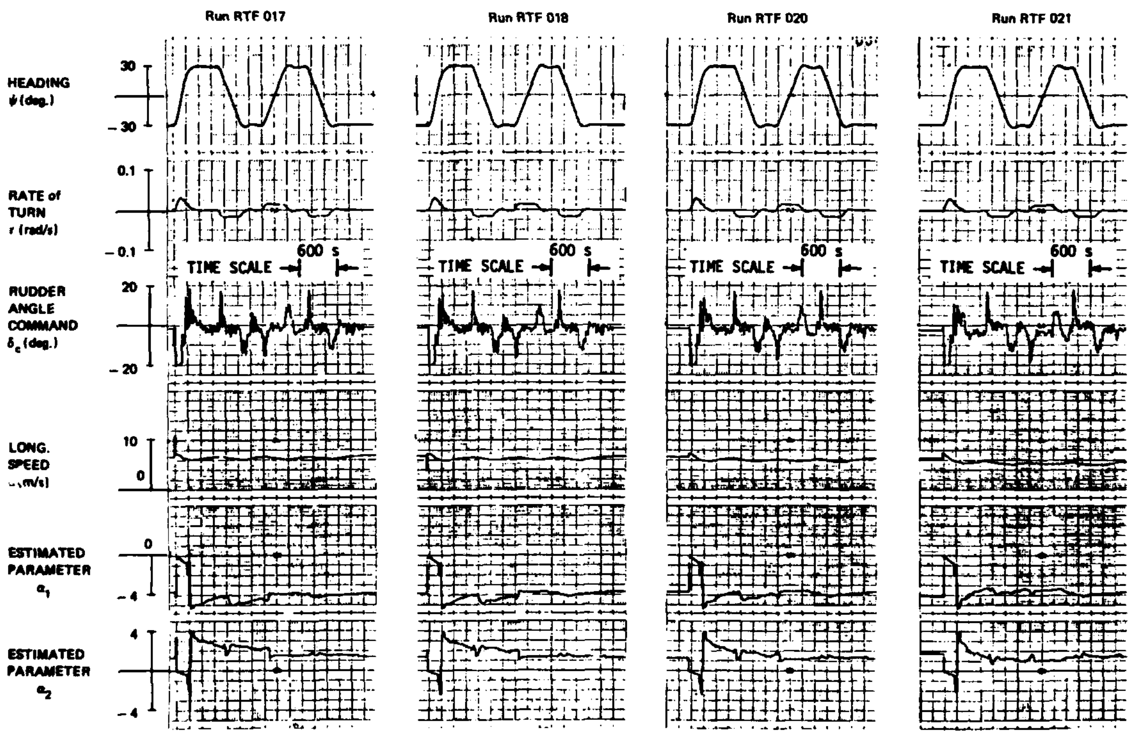

Figure 15 : Adaptive course changing of the supertanker (fully laden) in different stationary conditions.

to zero, i.e. by conditions of initial complete ignorance about the ships dynamics. Nevertheless, it can be observed that the course changing manoeuvres are very accurate in all different environmental conditions.

8. REFERENCES

(1) Ya. Z. Tsypkin, "Adaptation, training and self-organisation in automatic systems", Aut. Remote Control, Vol. 27, 1966.

(2) Y. Bar-Shalom, "Concepts and methods in stochastic control", in Control and Dynamic Systems ed. by C.T. Leondes, Academic Press, 1976, pp. 99 - 170.

(3) C.G. Källström and N.H. Norrbin, "Performance criteria for ship autopilots: an analysis of shipboard experiments", Symposium on Ship Steering Automatic Control, Genoa, Italy, June 1980.

(4) S. Motora and T. Koyama, "Some aspects of automatic steering of Ships", Japan Shipbuilding and Marine Engr., Vol. 3, No. 4, July 1968, pp. 5 - 18.

(5) J. v. Amerongen and H.R. van Nauta Lemke, "Criteria for optimum steering of ships", Symposium on Ship Steering Automatic Control, Genoa, Italy, June 1980.

(6) K.J. Åström, "Why use adaptive techniques for steering large tankers?" International Journal on Control, Vol. 32, 1980, 688 - 708.

(7) K.J. Åström and C.G. Källström, "Identification of ship steering dynamics", Automatica (12), 1976,pp. 9 - 22.

(8) A. Tiano and E. Volta, "Application of the identification techniques to the ship system", 7th IFAC Triennial World Congress, Helsinki 1978, Pergamon Press, pp. 1583 - 1588.

(9) C.G. Källström and K.J. Åström, "Experiences of system identification applied to ship steering dynamics", 5th IFAC Symposium on Identification and System Parameter Estimation, Darmstadt 1979, Pergamon Press.

(10) M. Cuneo, A. Tiano, A. Frosini, E. Mosca, P. Piazzesi, M. Sguanci, and P. Zappa, "Multivariable model identification of ship dynamics", 5th IFAC Symposium on Identification and System Parameter Estimation, Darmstadt 1979, Pergamon Press.

(11) K. Nomoto, "Response analysis of maneuverability and its application to ship design", Researches on the maneuverability of ships in Japan, Chapter 2, Soc. Nav. Arch., Japan, 50th Anniversary Series, 1966.

(12) M.I. Bech, "Some aspects of the stability of automatic course control of ships", J. of Mech. Engr. Sci., Vol. 14, No. 7, 1972.

(13) K.J. Åström, "Introduction to stochastic control theory", Academic Press, New York, 1970.

(14) R.E. Reid and P.F. Parent, " The application of linear regulator techniques to minimization of steering losses suffered by a ship in a

seaway", to appear in Journal of Dynamic Systems, Measurement and Control.

(15) K.J. Åström, "Design of fixed gain and adaptive autopilots based on the Nomoto model", Symposium on Ship Steering Automatic Control Genoa, Italy, June 1980.

(16) K. Ohtsu, M. Horigome, and G. Kitagawa, "A new ship's autopilot design through a stochastic model", Automatica, Vol. 15, 1979.

(17) K. Ohtsu, M. Horigome and G. Kitagawa, "A robust autopilot system against the various sea conditions", 3rd ISSOA Symp., Tokyo, 1979, North Holland Publ. Comp.

(18) H. Akaike, "A new look at the statistical model identification", IEEE Trans., AC-19, 1974, pp. 716 - 723.

(19) D.L. Kleinman, "On an iterative technique for Riccati equation computation", IEEE Trans., AC - 13, 1968, pp. 114 - 115.

(20) G.C. Källström, K.J. Åström, N.E. Thorell, J. Eriksson, and L. Sten: "Adaptive autopilots for tankers", Automatica, Vol. 15, 1979, pp. 241 - 254.

(21) F. Tosi and E. Verde, "Micrprocessor based adaptive autopilot: system development and sea trials", Symposium on ship steering automatic Control, Genoa, June 1980.

(22) A. Sugimoto and T. Kojima, "A new autopilot system with condition adaptivity", 5th ship Control Systems Symp., Annapolis, Md., 1978

(23) A.C. Schilling, "Economics for Autopilot Steering using an IBM System/7 Computer", 2nd IFAC/IFIP Symp. on Ship Operation Automation, Washington D.C., 1976.

(24) I.D. Landau, " Adaptive control - The model reference approach", Dekker, New York, 1979.

(25) J. van Amerongen and H.R. Van Nauta Lemke, "Optimum steering of ships with an adaptive autopilot", 5th Ship Control Systems Symposium, Annapolis, Md, 1978.

(26) J. van Amerongen and H.R, Van Nauta Lemke, "Experiences with a digital model reference adaptive autopilot", 3rd ISSOA Symp., Tokyo, 1979.

(27) K.J. Åström, "Self-tuning regulators - Design principles and applications", In Narendra & Monopoli: Applications of Adaptive Control, Academic Press, 1980.

(28) K.J. Åström and B Wittenmark, "On Self-tuning Regulators", Automatica, Vol.9 1973, pp. 185 - 199.

(29) D.W. Clarke and P.J. Gawthrop, "Self-tuning controller", Proc. I.E.E., Vol. 122, 1975, pp. 929 - 934.

(30) C.G. Källström, "Identification and adaptive control applied to ship steering", PhD Thesis, Department of Automatic Control, Lund Institute of Technology, Lund, Sweden, 1979.

(31) N. Mort and D.A. Linkens, "Self-tuning controllers for surface ship course and track keeping", Symposium on Ship Steering Automatic Control, Genoa, June 1980

(32) A.W.Brink, G. Baas, A. Tiano, E. Volta, "Adaptive automatic course keeping control of a supertanker and a containership. A simulation study", International Shipbuilding Progress, Vol. 26, n. 301, 1979, pp. 189 - 214.

(33) A.W.Brink and A. Tiano, "Self-tuning adaptive control of large ships in non-stationary conditions", International Shipbuilding Progress, Vol. 28, n. 323, 1981, pp. 162 - 178.

(34) R.E. Shannon, "Systems simulation - the art and science", Prentice - Hall, Inc., Englewood Cliffs, New Jersey, 1975.

(35) B.P Zeigler, "Theory of modelling and simulation ", John Wiley & Sons, New York, 1976.

(36) J.S. Annino and E.C. Russell, "The ten most frequent causes of simulation analysis failure - and how to avoid them!", Simulation, no. 60, June 1979.

(37) M.I. Bech and L. Wagner - Smitt, "Analogue Simulation of ship manoeuvres", Report Hy - 14, Hydro-og Aerodynamisk lab, Lyngby, Denmark, 1969.

(38) M.J. Morgan, "Dynamic positioning of offshore vessels", The Petroleum Publishing Co., Tulsa, USA, 1978.

COMPARISON OF AUTOMATIC STEERING PERFORMANCE OF A VLCC IN A SEAWAY RESULTING FROM APPLICATION OF LQG AND CLASSICAL CONTROL SYSTEM DESIGN TECHNIQUES

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Abstract

The problem of automatic steering control of a ship during course-keeping in a seaway is addressed in terms of controllability and propulsion losses. The steering performance of a 1,085-ft long tanker (250,000 dwt) under full load and ballast conditions, under the action of steering controllers resulting from the application of both linear-quadratic Gaussian (LQG) and classical control system design techniques to minimization of a performance criterion commonly believed to be representative of propulsion losses, is examined using time domain simulation and frequency domain analysis techniques. The ship model used is based on hydrodyanmic data deriving from captive scale model tests. The performance of these controllers is also compared to that resulting from existing conventional autopilots. On the basis of the results presented, the validity of a stochastic approach to controller design in the environment afforded by the seaway is re-examined and the implications for new autopilot design appropriate to this type of ship discussed.

1. INTRODUCTION

For several years the U.S. Maritime Administration has been investigating the means of improvement in the performance of modern ship types under automatic steering control [1-5]. In addition to the controllability of such ships, the economics associated with ship operations have necessitated an examination of the propulsion losses associated with the motion of an automatically steered ship in a seaway. The problem of steering control of a very large crude carrier (VLCC) is particularly demanding due to the inherent dynamic instability of the ship when fully loaded. The advent of the microprocessor allows implementation of rather sophisticated real-time control algorithms, and there have been several recently proposed autopilot designs for commerical shipping employing these e.g. [6-8]. This paper presents results obtained during an ongoing research effort sponsored by the U.S. Maritime Administration which is directed specifically to identification and minimization of propulsion losses related to ship steering. The steering performance of a 1085 ft long tanker (250,000 dwt) in a seaway under full load and ballast conditions, under the action of steering controllers resulting from the application of both Linear-Quadratic-Gaussian (LQG) and classical control system design techniques to minimization of a performance criterion commonly believed to be representative of propulsion losses, is examined using time domain simulation and frequency domain analysis techniques. The performance of these controllers is also compared to that resulting from existing conventional autopilots.

Much of the recent research effort into automatic ship steering control has been directed to attempted minimization of propulsion losses related to steering by adaptive controllers based on stochastic, model reference, heuristic, and self-optimalizing techniques [6-12]. Astrom [13], however, recently demonstrated the use of linear quadratic regulator techniques in this problem. By this means he showed substantial simplification of the implementation of an adaptive autopilot design could be made. In that work [13], however, a fairly simple model of the ship and seaway disturbances was used which, although enabling considerable insight into the system dynamics, did not allow determination of the resulting propulsion losses from application of the linear regulator design. This paper specifically extends the recent results of Astrom [13] in LQG design of ship steering controllers by examining the sensitivity of controllers designed to a common form of quadratic performance criterion to seaway disturbances, and by assessing the effect of variation of the relative weightings in the criterion on the resulting system performance. The ship model used is based on hydrodynamic data deriving from captive scale model tests [14]. The system model has not taken account of the small signal nonlinearities of the "two-loop" steering gear common on many such ships [5,15]. It is, however, strictly applicable to the "single-loop" steering system alternative also in use [5,16]. The results relate to open-seas, steady-state course-keeping in deep water. On the basis of the results presented, the validity of a stochastic approach to controller design in the environment afforded by the seaway is re-examined and the implications for new autopilot design appropriate to this type of ship discussed.

2. SYSTEM DYNAMICS

2.1 Ship Dynamics

The principal characteristics of the VLCC in both full-load and ballast conditions is shown in Table 1. Figure 1 shows the coordinate system used to define ship motions. Longitudinal and transverse horizontal axes of the ship are represented by the x- and y-axes with origin fixed at the center of gravity. By reference to these body axes, the equations of motion of the ship in the horizontal plane can be written in the form

$$\begin{aligned} I_z \dot{r} &= N && \text{(yaw)} \\ m(\dot{v} + ur) &= Y && \text{(sway)} \qquad (1) \\ m(\dot{u} - vr) &= X && \text{(surge)} \end{aligned}$$

where m is the ship's mass and I_z its inertia about the yaw axis and where N, Y, and X represent total hydrodynamic terms generated by ship motions, rudder, and propeller [14]. In studying the basic steered motion of a surface ship, pitch and heave motions can be neglected [17]. For full form, low speed vessels such as tankers, roll effects on steering and vice-versa are generally small and can usually be neglected in steering performance analysis [14].

The formulation of these equations is well known [17-19]. Since it has been shown that, while nonlinear effects are necessary for prediction of ship motions in maneuvers, linear treatment is effective for prediction of lateral ship motions in waves during course-keeping [17], the second and higher order hydrodynamic terms are not included in the model used in the analysis of course-keeping controllers. The coefficients used in this work are based on captive scale model tests [14]. The perturbation equations of the ship about an operating point in the lateral plane in waves may then be written in a state-space vector form

Table 1 Principal Particulars: 250,000 DWT Tanker

Length between Perpendiculars:--1085 ft.
Beam---------------------------170 ft.
Depth---------------------------84 ft.
Draft---------------------------65 ft., 5 3/4 in. (full load)
Displacement----------------285,944 tons (full load)

	Full Load Condition in Deep Water	Ballast Condition in Deep Water
Length Beam Ratio, L/B	6.380	6.380
Beam Draft Ratio, B/H	2.600	5.460
Block Coefficient, CB	0.830	0.757
Rudder Area Ratio, AR	0.019	0.030
Trim, TRIM	0.000	0.009

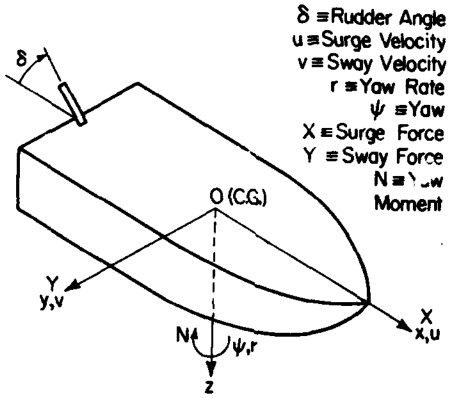

Figure 1 Ship Coordinate System

Table 2 System Matrices for 250,000 DWT Tanker at 15 Knots

Full Load:

$$A = \begin{bmatrix} -0.1306 \times 10^{-1} & -0.8639 \times 10^{+1} & 0 \\ -0.9309 \times 10^{-4} & -0.4314 \times 10^{-1} & 0 \\ 0 & 1 & 0 \end{bmatrix}$$

$$B = \begin{bmatrix} 0.7037 \times 10^{-1} \\ -0.4592 \times 10^{-3} \\ 0 \end{bmatrix} \qquad D = \begin{bmatrix} 0.272 \times 10^{-7} & 0 & 0 \\ 0 & 0.3651 \times 10^{-12} & 0 \\ 0 & 0 & 0 \end{bmatrix}$$

Open-Loop Eigenvalues
(includes effect of rudder servo-dynamics):

0, -0.06020, 0.00400, -0.3333

Ballast:

$$A = \begin{bmatrix} -0.2019 \times 10^{-1} & -0.1199 \times 10^{+2} & 0 \\ -0.3679 \times 10^{-4} & -0.3996 \times 10^{-1} & 0 \\ 0 & 1 & 0 \end{bmatrix}$$

$$B = \begin{bmatrix} 0.1415 \times 10^{0} \\ -0.9786 \times 10^{-3} \\ 0 \end{bmatrix} \quad D = \begin{bmatrix} 0.7670 \times 10^{-7} & -0.2212 \times 10^{-10} & 0 \\ -0.2212 \times 10^{-10} & 0.1107 \times 10^{-11} & 0 \\ 0 & 0 & 0 \end{bmatrix}$$

Open-Loop Eigenvalues
(includes effect of rudder servo-dynamics):

0, -0.05328, -0.0069, -0.333

$$\dot{X} = AX + Bu + DW \tag{2}$$

where X is the state vector $(v, r, \psi)^T$, u the control vector, and W the disturbance vector $(Y_D, N_D, 0)^T$, where Y_D and N_D refer to the external disturbances produced by the seaway. For the 250,000 DWT tanker at an operating speed of 15 knots in the fully loaded condition the system matrices of Eq. (2) are as shown in Table 2. The system matrices for the ship in ballast at the same speed are shown also in Table 2. An examination of the dynamics shows that the ship is open-loop unstable fully-loaded and stable in ballast, as is indicated by the eigenvalues shown also in Table 2.

The surge equation may be written to include the effects of hydrodynamic forces, propeller force, and external forces which also result in propulsion losses, as [14]

$$(m - C_4)\,\dot{u} = C_0 U^2 + (m + C_1)\,vr + C_2\,v^2 + C_3\,U^2\,\delta^2 + X_p + X_{extern} \tag{3}$$

where C_4 is the derivative of hydrodynamic force component in the x-axis (longitudinal axis) direction with respect to surge acceleration; $\dot{u}$ is the surge acceleration; U is ship's speed; C_o is drag coefficient; C_1 is second derivative of hydrodynamic force component in x-axis direction with respect to sideslip (sway) velocity and yaw angular velocity; C_2 is second derivative of hydrodynamic force component in x-axis direction with respect to sideslip velocity; C_3 is second derivative of hydrodynamic force component in x-axis direction with respect to rudder angle; and where X_p is propeller thrust and X_{extern} external forces acting in the x-direction.

For the small excursions from steady advance considered, the coupled yaw/sway equations describing the linear hull dynamics may be considered to be decoupled from the surge equation which contains only second order terms in yaw rate, sway velocity, and rudder angle. Equation (1) in yaw and sway describes the dynamics to an input of rudder angle or external disturbances. The resulting sway velocity and yaw rate, along with rudder angle can be considered as inputs to the surge equation for purposes of steering analysis. With the further assumptions of constant thrust deduction factor and wake fraction [20], the surge force related to steering is

$$\Delta X = (m + C_1)\ vr + C_2 v^2 + C_3\ U^2\ \delta^2 \qquad (4)$$

This may be used to determine the propulsion losses related to steering in both calm water and waves [21]. The nonlinear force coefficients used derive from captive model tests [14].

2.2 Steering Gear Dynamics

The steering system is an electrohydraulic servomechanism driving a hydraulic pump which discharges oil to a hydraulic actuator which in turn rotates the rudder through a tiller arrangement [6]. In general, there are small signal nonlinearities in the servo driving the pump and large signal nonlinearities in the pump itself [6,22]. For purposes of linear system analysis, the small signal nonlinearities may be neglected [22]. Linear operation of the pump is also a reasonable assumption during normal course-keeping operation of the ship [6,7] if good control is exercised. The validity of this assumption may be checked in <u>a posteriori</u> performance evaluation of the system. On this basis, the steering gear dynamics can be represented by

$$\delta/\delta_c = \frac{1}{(1 + \tau s)} \qquad (5)$$

where δ is rudder angle, δ_c is rudder command and τ is the time constant of the system equal to 3 seconds.

2.3 Seaway Dynamics

The model of the seaway environment encountered by the ship is based on observed data for Beaufort 8, i.e. fresh gale, conditions [23] fitted to the International Ship Structure Congress (ISSC) seaway energy spectrum [24] given by

$$G_{\xi\xi}(\omega) = [(172.8\ H_s^2)/(T_s^4\ \omega^5)]\ e^{-691/(T_s^4\omega^4)} \qquad (6)$$

where H_s and T_s are, respectively, observed significant wave height and period and where $G_{\xi\xi}(\omega)$ is the spectral density ordinate defined such that the mean square value of wave amplitude in a frequency interval $\Delta\omega$ is

$$1/2\ a^2\ (\omega) = G_{\xi\xi}(\omega)\ \Delta\omega \qquad (7)$$

Beaufort 8 conditions may be reasonably considered the maximum weather conditions under which running cost considerations are of importance. The relative speed and direction of the ship to the waves causes a shift in the frequencies of the encountered waves. In terms of wave circular frequency ω, the encounter frequency ω_e is

$$\omega_e = \omega - \omega^2 U (\cos \beta)/g \qquad (8)$$

where β is the ship's heading relative to the wave direction [25]. The resultant sea spectra encountered by the ship at its cruise speed of 15 knots in these conditions transformed from the spectra of Eq. (8) as a function of encounter frequency for various encounter angles are shown in Fig. 2. An encounter angle $\beta = 0°$ represents following seas, $\beta = 180°$ head seas [25]. From the form of Eq. (8), it may be seen that $\omega_e(\omega)$ is not unimodal and results in negative values for some range of wave frequencies for $0 < \beta < 90°$, i.e. following to beam seas. This is reflected in the form of the transformed spectra of Fig. 2a, where the spectra for negative frequencies have been folded onto the positive axis, as is necessitated by the logarithmic abscissa of the figures. The spectrum for any encounter angle is continuous through zero frequency. There are fairly sharp resonances in the spectra for seas aft of the beam around 0.3 to 0.6 radians per second, but otherwise the magnitudes are lower than for those energy spectra for seas forward of the beam. As is shown in Fig. 2b for beam seas ($\beta = 90°$), the projection of the speed of the ship along the wave direction of propagation is equal to zero and the original spectrum is obtained. The greatest seaway energy occurs around 0.6 radians per second. For seas forward of the beam, ($\beta > 90°$), the spectrum is shifted towards the high frequencies and slightly attenuated.

The disturbances to the ship/steering system resulting from the seaway excitation may be approximated by integration of wave pressure on the local sections along the longitudinal axis of the ship's hull on the basis of the Froude-Kriloff theory [25]. The sway force and yaw moment exerted by a component wave on the ship may be computed by numerical integration [6] to achieve a representation of the disturbance to the ship which is a function of wavelength, wave amplitude, wave direction (relative to the nominal ship heading), the hull section parameters and time [6]. This formulation is primarily directed to time-domain simulation studies, due to the difficulty of a closed-form analytical solution to the evaluation of steering performance in the seaway [6]. A frequency domain representation of the disturbances, however, is also possible. Figures 3 and 4 show magnitude and phase of yaw moment/wave amplitude and sway force/wave amplitude, respectively, as a function of wave frequency for following through bow seaway encounter angles (β = 30, 45, 60, 90, 120, and 150 degrees), for the full-load condition. Figures 3 and 4 may be used in conjunction with the shifted seaway energy spectra of Fig. 2 to determine the frequency range of seaway disturbances suffered by the system. The ballast disturbance/wave amplitude characteristics are similar [26].

3. PERFORMANCE UNDER CONVENTIONAL AUTOPILOT CONTROL

3.1 The Conventional Autopilot

In conventional steering systems the automatic steering system control loop is closed through the gyrocompass and autopilot. The measured heading signal from the gyrocompass is compared with the desired heading and the error is input to the controller. Typically the controller output drives the rudder servo through interface equipment as described in [6,22]. Traditionally the steering system has been treated as essentially a one-input (heading command), one-output (ship heading as measured by gyrocompass), unity gain feedback system controlled by a fixed-structure control system [22]. The standard form of such autopilots is based on proportional-plus-integral-plus derivative (PID) control, in an

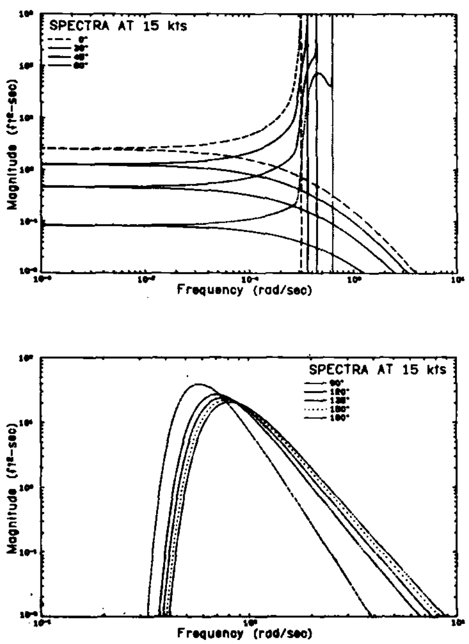

Figure 2 Sea Spectra for Different Encounter Angles: Beaufort 8, 15 Knots

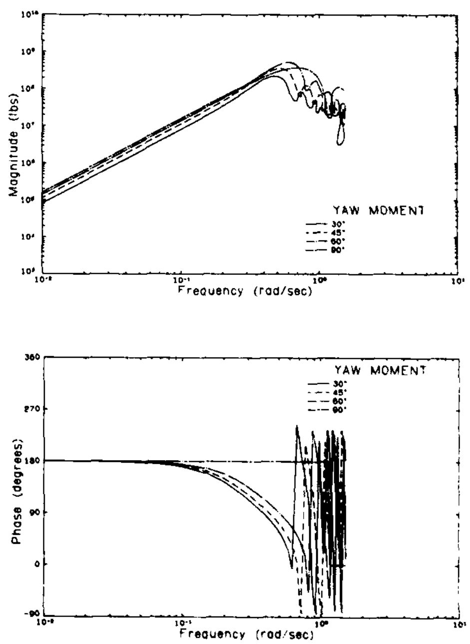

Figure 3 Frequency Response of Yaw Moment/Wave Amplitude: 250,000 DWT Tanker, Full Load

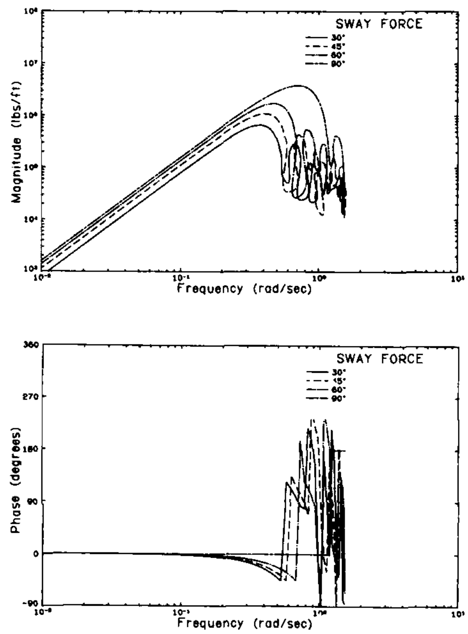

Figure 4 Frequency Response of Sway Force/Wave Amplitude 250,000 DWT Tanker, Full Load

analog computer implementation [6]. No provision for automatic adaptivity of the controller to speed, load, or seaway exists. Instead, it relies on adjustment of the controller parameters by the operator [5,27], but the design in many cases precedes the modern generation of ships.

An autopilot commonly found aboard merchant ships [5,27] has the following characteristics. The control law may be described by:

$$\delta_c = K_1[K_2(1 + (\tau_1 s/(1 + \tau_2 s)^2) + (1/(\tau_3 s))]\psi \qquad (9)$$

where δ_c is the ordered rudder angle, ψ the heading error, and where the gains are determined by operator interaction in the following manner:

(1) "Weather adjust" gain

$$K_1 = \begin{cases} 1/3 \text{ within "weather adjust" zone} \\ 1 \text{ outside "weather adjust" zone} \end{cases}$$

The weather adjust zone is continuously variable from 0.3 to 5.0 degrees heading error (panel is marked "0" to "5").

(2) "Rudder multiplier" gain

K_2 = 1 to 3, continuously adjustable (panel control)

The time constants τ_1, τ_2, τ_3 are preset from a range of possible discrete values for the specific ship, with the exception that τ_1 may be reduced by a "Rate multiplier" control by a factor of 2 by operator action.

The basis for controller action at specified operational conditions, then, is determined by specification by the operator of "Weather adjust" zone width, "Rudder multiplier," and "Rate multiplier." The use of integral control is optional. It is used for correction of steady-state errors resulting from system asymmetric or bias effects [22]. By proper selection of the effective integral control time constant, the dynamic characteristics of the controller are largely unaffected. For the purpose of this analysis, which addresses dynamic effects, no consideration is given to integral control effects. The effects of gyrocompass dynamics and kinematics [6,7] are also neglected.

3.2 Sensitivity Analysis

A qualitative examination of steering performance of the ship in a seaway can be made using frequency domain analysis techniques [28]. This method offers considerable insight into the problem, since the bandwidths of interest for the controllers in different conditions may be established using the seaway energy spectra of Fig. 2 and the seaway disturbances to wave amplitude frequency responses shown in Figs. 3 and 4. These may be related to frequency domain representations of system performance to disturbances. Conventional autopilot performance for the ship in the fully loaded condition is discussed on this basis first, followed by an examination of its performance as evaluated by time domain simulation studies.

Figures 2 through 4 allow a determination of the frequency range of seaway disturbances to which the ship is subjected. From these it may be inferred that the seaway disturbances on this ship under the conditions examined exhibit bandwidths from 0.35 to 1.00 radians per second for beam seas, 0.4 to 1.0 radians per second for seas forward of the beam and -1.0 to 0.4 radians per second for following/quartering seas. The performance of the conventional autopilot may be assessed in terms of controllability and added resistance. The settings for the controller time constants correspond to those found on similar ship types and are representative of large tanker settings. The effect of operator control of weather adjust zone, rudder multiplier gain, and rate multiplier is examined by analyzing three different setting conditions ranging from low to high bandwidth control (i.e. "slack" to "tight" attempted heading control). These are designated as controllers CV1', CV2 and CV3, respectively. In terms of the parameters of Eq. (9), these correspond to the following controller settings:

	K_1K_2	τ_1 (secs)	τ_2 (secs)
CV1'	0.33	75	15
CV2	1.0	100	15
CV3	2.0	150	15

System sensitivity to seaway disturbances is examined by consideration of yaw moment [28]. The closed-loop yaw deviation (i.e. heading error) to yaw moment is shown in the magnitude-frequency response of Fig. 5 for each controller. The high bandwidth controller CV3 appears to offer attenuation of low frequency disturbances while suffering from a resonance around 0.07 radians per second. The low bandwidth controller CV1' suffers apparent degradation in performance at the low frequencies. Under the conditions being examined, controller CV2 is probably the best choice for controllability. It is very probable, however, that the operator would select control settings represented by controller CV3, in an effort to improve the ship's heading performance by increased bandwidth [5,27]. Further, although it is not the subject of this paper, the range of controls available to the operator is such that the ship/steering system can lose directional stability in either the low or high gain control ranges. The settings simulated in this study have been carefully selected to ensure closed-loop stability of the control system. The operator has no such knowledge available to him, and relies on trial and error from examination of the resulting heading error displayed to him on the ship's bridge [5,27].

It is possible to assess the sensitivity of added resistance due to steering from Eq. (4). Neglecting the term in (sway velocity)^2, which may be shown to be generally small relative to the other two terms of Eq. (4), the added resistance may be expressed in normalized form as

$$\Delta R' = -\lambda'' \; v\dot{r} + \delta^2 \tag{10}$$

where

$$\lambda'' = - [(m + C_1)/C_3U^2] \tag{11}$$

The spectral density of the normalized total added resistance (Eq. (10)) to yaw moment spectral density is shown in Fig. 6 [27]. An analysis of Fig. 6, in conjunction with the seaway disturbance frequency information of Figs. 2 and 3, enables a qualitative evaluation to be made of the relative efficiency of the controllers in terms of energy losses. The phase of the frequency response of Fig. 6 is important in addition to its magnitude in determination of losses due to yaw/sway [25]. It may be shown that the ship suffers an increase in added resistance due to yaw/sway at all but high frequencies [27]. While controllers CV1' and CV3 appear to perform worse, overall, in terms of yaw/sway added resistance suffered by the ship, than controller CV2, no controller is able to effect a reduction in the added resistance of the uncontrolled ship. All controllers, however, in fact result in an increased penalty in added resistance due to their respective resulting induced rudder drag in attempted attenuation of heading error [27]. In terms of net added resistance controller CV2 appears to result in the lowest overall penalty. Controller CV1' pays a penalty for its reduced controllability performance, which results in greater yaw/sway activity [27].

3.3 Time Domain Simulation Studies

A quantitative evaluation of performance is provided by digital time-domain simulation techniques [6,7]. A Beaufort 8 seaway representation is effected by driving the seaway disturbances of sway force and yaw moment by an irregular seaway model composed of the summation of 25 discrete components of the energy spectrum of Eqs. (6) and (7) randomly mixed. The irregular seaway model is obtained by using random phase angles, from a uniform distribution between $-\pi$ and π for each wave component of the discretized spectrum [6]. The integration of the state equations of the system (Eq. (2)) is accomplished using a fourth order form of Runge-Kutta numerical integration method. The large-scale nonlinearity of the steering pump saturation [6], is modeled in the simulation. Simulation studies show that one run of the program for the equivalent of 2,000 seconds of real time with an arbitrary set of initial random phase angles achieves an adequate representation of steady-state performance in an irregular seaway.

The qualitative frequency domain analyses are essentially borne out by the time-domain simulation results of Table 3 and Fig. 7. The steady-state

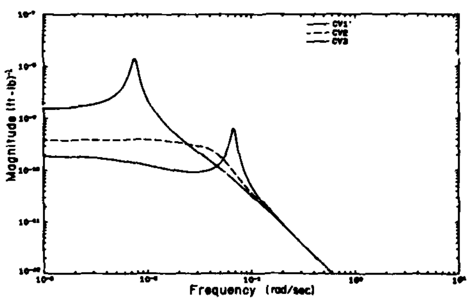

Figure 5 Frequency Response of Yaw/Yaw Moment: 250,000 DWT Tanker, Full Load, Conventional Autopilot

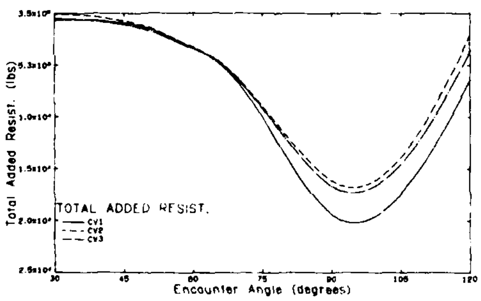

Figure 7 Total Added Resistance due to Steering: 250,000 DWT Tanker, Full Load, Conventional Autopilot

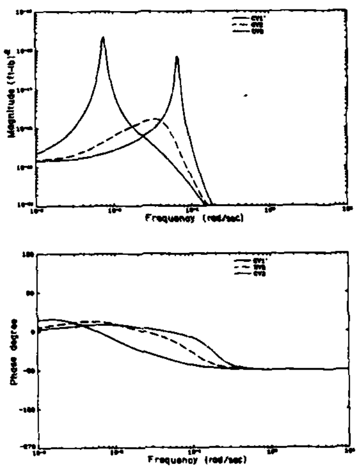

Figure 6 Spectral Density of Total Added Resistance/Spectrum of Yaw Moment: 250,000 DWT Tanker, Full Load, Conventional Autopilot

Table 3 Steering Performance of Conventional Autopilot
250 000 DWT Tanker: Full Load, 15 Knots, Beaufort 8

Encounter Angle (deg)	Sway Velocity (ft/s)	Yaw Rate (deg/s)	Yaw (deg)	Rudder Angle (deg)	Rudder Rate (deg/s)	Mean Added Resistance (lb) Yaw/Sway	Sway^2	Rudder	Total
CV1'									
30	0.12	0.03	1.27	0.42	0.01	910	70	60	1,040
45	0.16	0.05	0.83	0.26	0.01	1,130	120	20	1,280
60	0.28	0.07	0.59	0.18	0.01	3,270	360	10	3,650
90	0.58	0.05	3.00	0.93	0.01	17,640	1,510	300	19,460
120	0.29	0.05	3.44	1.07	0.01	5,850	310	400	6,640
CV2									
30	0.08	0.03	0.15	0.34	0.04	340	30	40	410
45	0.15	0.05	0.16	0.29	0.05	30	110	30	1,070
60	0.28	0.06	0.16	0.19	0.06	3,190	350	10	3,560
90	0.54	0.04	0.32	0.65	0.03	14,770	1,320	150	16,240
120	0.18	0.05	0.38	0.81	0.04	2,100	140	230	2,480
CV3									
30	0.09	0.03	0.13	1.12	0.12	400	30	430	870
45	0.15	0.05	0.14	0.95	0.15	990	110	320	1,420
60	0.28	0.07	0.15	0.54	0.16	3,270	360	100	3,680
90	0.54	0.04	0.17	1.40	0.11	14,740	1,320	690	16,750
120	0.17	0.05	0.22	2.12	0.15	2,190	140	1,560	3,900

performance of the ship/steering system for each of the controllers resulting from simulation studies is presented for the full-load condition at 15 knots in Table 3. R.M.S. motion of the ship in sway velocity, yaw rate, and yaw deviation, R.M.S. values of rudder angle and rudder rate, and the components of mean added resistance due to steering are shown for relative seaway directions progressing from following (β = 30 degrees) through bow (β = 120 degrees) seas. Figure 7 shows a comparison of the resulting added resistance suffered by the ship/steering controllers as a function of encounter angle. It may be seen that controller CV2 results in the lowest added resistance. The low bandwidth controller CV1', due to its resulting increased yaw, results in greater net losses than does the high bandwidth controller CV3. Over the limited number of conditions examined by time-domain simulation studies the maximum net loss found occurs at 90 degrees encounter angle, and amounts to some 4.3 percent of normal full power [27]. The form of the sensitivity responses (Fig. 6) indicates, however, that a potentially greater loss than this can result from control settings corresponding to controllers CV1' and CV3 at some seaway encounter angles not considered in simulation studies.

These results, in addition to evaluating the performance of a conventional autopilot, also demonstrate the operator's difficulty in setting the autopilot for purposes of either controllability or propulsion economy, whatever his intent. The result of automatic steering control by the conventional autopilot examined is, in general, to increase the net propulsion losses related to yawing/swaying and steering control actions [27]. This is due mainly to the increased rudder induced drag resulting from controller actions. There appears to be little basis for the conventional autopilot to be set to minimize energy losses by the operator. The need for a steering controller designed to a performance criterion representative of steering related propulsion losses, with adaptivity to speed, load or seaway changes seem, on the basis of the results presented, would appear to be well established.

4. LINEAR QUADRATIC FORMULATION

4.1 Performance Criterion

The accepted requirements for the steering control system are that it should guarantee stability over a wide range of operating conditions and provide good controllability while minimizing propulsion losses related to steering during course-keeping. There is less than universal agreement among researchers on specification of a performance criterion applicable to open-seas steering control [11-13,20,21,29-35]. A form of criterion based on an approximation for added resistance per unit distance due to steering by Norrbin [20] has been used by several researchers including Astrom [13,36], and it is used as a basis for controller design in this paper. It is given by

$$J = \lim_{T\to\infty} \frac{1}{2T} \int_0^T [\lambda\, \psi^2 + \delta^2]\, dt \qquad (12)$$

The value of λ determines the relative penalties placed on yaw deviations and rudder angle. Based on an approximation for added resistance per unit distance due to steering [6,7,21], values of λ may be calculated from the nonlinear force coefficients of Eq. (4), which are dependent on ship speed and loading and which are also dependent on the desired system bandwidth [6,7,21]. Values of λ may be chosen which penalize yaw deviations to a greater or lesser amount. The values of λ corresponding to each case for the 250,000 DWT tanker are shown in Table 4 for a ship speed of 15 knots for the full-load and ballast conditons. These compare with the value of λ = 10 for a tanker used by Astrom [36] based on the work of Norrbin [20].

Table 4 LQG Controller Characteristics

Full Load:

LQG	Weighting Coefficient,	Feedback Gain $u^* = -KX$		
		k_v	k_r	k_ψ
1	29.5	-0.16	100.40	5.43
3	7.4	-0.15	65.81	2.72

Closed Loop System Eigenvalues:

LQG	Eigenvalues
1	-0.03898, -0.03748 $\pm j$ 0.01855
3	-0.05704, -0.01988 $\pm j$ 0.014217

Ballast:

LQG	Weighting Coefficient,	Feedback Gain, $u^* = -KX$		
		k_v	k_r	k_ψ
2	15.8	-0.03	61.45	3.98
4	3.9	-0.03	38.74	1.99

Closed-Loop System Eigenvalues:

LQG	Eigenvalues
2	-0.02680, -0.04892 $\pm j$ 0.03618
4	-0.03651, -0.03267 $\mp j$ 0.01706

4.2 Regulator Design

With the linear model of the ship (Eq. (2)), controllers to minimize the performance criterion of Eq. (12) at each operating condition defined may be designed using optimal quadratic regulator techniques [26]. If it is initially assumed that perfect knowledge of the states of the system is available, and with the further assumption of white process noise, as used in [13], the optimal LQG controllers, i.e. which minimize the expected value of the criterion of Eq. (12), are identical to the deterministic controller [37]. The conditions for a unique solution are well known [37]. The relative weighting penalty, resulting control laws, and closed-loop system eigenvalues are shown in Table 4. The study concerns a dynamic analysis only of the steering problem in waves. In general, an integral control term would also be used to account for static or slowly-varying disturbances or modeling discrepancies [25] in the design implementation of the regulator. It may be seen that the gains applied to sway velocity, v, in all of the controllers are small and of opposite sign to the other state feedback gains.

4.3 Linear Regulator Performance

The resulting controllers were subjected to sensitivity and simulation studies to determine their relative performance in controllability and propulsion economy, assuming, as in their design, perfect knowledge of the system states [26]. The results for full-load and ballast conditions are summarized in Figs. 8 through 11 and Table 5. They show that the LQG controllers designed to the lower weighting penalties on yaw deviation enjoy a clear advantage over those designed to the higher yaw penalties in overall performance in both loading conditions. Excellent controllability is afforded by either of the controllers at each of the loading conditions. In providing this control, however, LQG1 (full-load) and LQG2 (ballast) exhibit considerably higher rudder angles and rates than do, respectively, LQG3 and LQG4. While this increased rudder action is apparently unable to reduce yaw/sway added resistance of the ship, it does result in increased rudder induced drag, as was also predicted by the frequency domain analysis [26]. In terms of overall performance, steering gear wear is also a consideration [22]. Examination of rudder rate, which is an direct measure of pump stroke [6], shows consistently greater steering gear activity resulting from use of the higher yaw penalty controllers. Very few instances of pump-stroke saturation are evidenced by the simulation studies, justifying the linear model used for the steering gear (Eq. (5)) for frequency domain analyses purposes, and also indicating good controller design in terms of controllability.

In the full-load condition, the net effect of the higher yaw penalty controller is to result in a mean added resistance due to steering of some 5.52 percent of normal full power in beam seas [26]. The lower yaw penalty controller, LQG3, affords a net reduction in losses under these conditons of 0.75 percent of full power. The losses due to steering in the ballast condition are not as great, especially in beam seas where the losses resulting from the higher penalty controller, LQG2, amount to 1.96 percent of full power (β = 60 degrees). The lower penalty controller LQG4 results in a net savings of 0.79 percent of full power under these conditons with no degradation in yaw performance (Table 5).

A comparison of the performance of these linear regulators for the full-load condition, however, with the performance of the conventional autopilot under the same conditons (Table 3), shows the former to be outperformed by the latter in almost all cases simulated. An examination of the results shows that it is the much higher R.M.S. rudder angle and resulting induced rudder drag of the LQG controllers which results in the poorer propulsion efficiency of these controllers while achieving little improvement, in general, in controllability (i.e. yaw deviation performance) over, for example, conventional controller settings corresponding to CV2 or CV3. The LQG controllers designed on the basis of a white noise disturbance environment pass frequencies higher than necessary in the actual seaway disturbance environment, causing higher rudder activity than desirable for minimization of added resistance. This effect may be attempted to be alleviated by the introduction of a Kalman filter to the system [13].

4.4 Kalman Filter Design

Initial linear regulator design [26] was conducted with the assumption of perfect state knowledge. The system states may be classified into those which are normally measured and those which are not. Measurements of sway velocity are normally not available on merchant ships. The system, however, is completely observable from the yaw deviation signal (measured from the gyrocompass). It is therefore possible to obtain estimates of all states using a Kalman filter or stochastic observer. Astrom [13] has demonstrated the design of a simple Kalman filter for LQG control on the basis of white seaway disturbance noise, where the filter gain increases with increasing sea state. In [12] a higher order

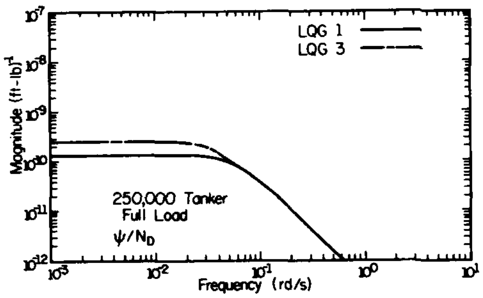

Figure 8 Frequency Response of Yaw/Yaw Moment: 250,000 DWT Tanker, Full Load, Controllers LQG 1 and LQG 3

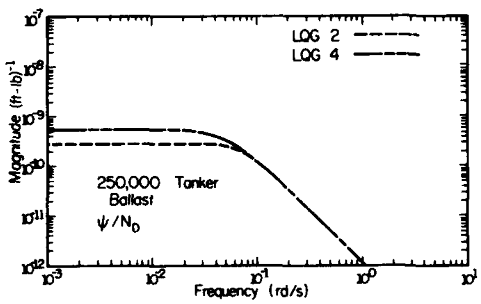

Figure 9 Frequency Response of Yaw/Yaw Moment: 250,000 DWT Tanker, Ballast, Controllers LQG 2 and LQG 4

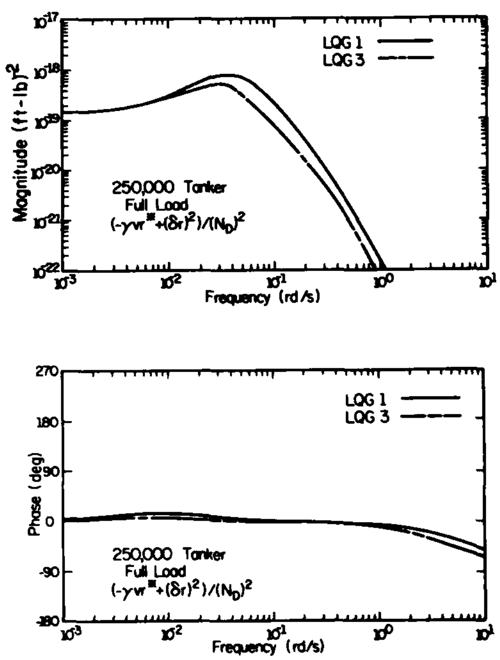

Figure 10 Spectral Density of Total Added Resistance/Spectrum of Yaw Moment: 250,000 DWT Tanker, Full Load, Controllers LQG 1 and LQG 3

Table 5 Steering Performance of "Perfect" Linear Regulators
250,000 DWT Tanker, 15 Knots, Beaufort 8

Encounter Angle (Deg)	Sway Velocity (ft/s)	Yaw Rate (Deg/s)	Yaw (deg)	Rudder Angle (Deg)	Rudder Rate (Deg/s)	Mean Added Resistance (lb_f) Yaw/Sway	Sway^2	Rudder	Total
Full Load LQG No. 1:									
30	0.09	0.03	0.11	2.53	0.83	460	30	2200	2690
45	0.16	0.05	0.14	3.93	1.35	1230	110	5290	6630
60	0.28	0.07	0.16	5.26	1.67	3810	360	9500	13670
90	0.55	0.05	0.09	4.79	1.85	15940	1360	7870	25160
120	0.17	0.05	0.10	2.59	1.40	2150	130	2310	4590
LQG No. 3:									
30	0.09	0.03	0.13	1.68	0.54	410	30	970	1410
45	0.16	0.05	0.14	2.63	0.97	1140	110	2370	3620
60	0.28	0.07	0.16	3.62	1.40	3650	360	4510	8520
90	0.55	0.04	0.11	3.73	1.69	15640	1350	4770	21750
120	0.17	0.05	0.14	1.87	1.09	2090	130	1210	3430
Ballast LQG No. 2									
30	0.13	0.05	0.17	2.44	0.80	460	160	1530	2150
45	0.25	0.09	0.25	3.85	1.35	1170	270	3560	5000
60	0.46	0.13	0.30	4.95	1.65	2520	670	5790	8970
90	0.94	0.02	0.04	1.31	0.80	3790	2460	580	6840
120	0.26	0.08	0.17	2.32	1.36	1480	290	1450	3220
LQG No. 4									
30	0.14	0.05	0.18	1.49	0.49	400	160	680	1240
45	0.25	0.08	0.26	2.36	0.88	1040	270	1440	2750
60	0.46	0.13	0.31	3.07	1.29	2360	670	2330	5350
90	0.94	0.02	0.05	1.01	0.62	3710	2460	420	6590
120	0.26	0.08	0.21	1.50	0.94	1420	290	720	2430

disturbance model was apparently used to include coloring in the seaway disturbance model in an adaptive control implementation.

Astrom [13] has argued for the modeling of the seaway disturbances as white noise in linear regulator and Kalman filter design on the basis of the low pass nature of a tanker. At the same time the acknowledgment that such a model did not describe low encounter frequency disturbance such as might be experienced by a tanker in following seas was made [13]. Since it has been shown that the bulk of the propulsion losses occur in beam seas for the tanker at its cruise speed in Beaufort 8 conditons at both full-load and ballast, Kalman filters were designed on this basis, i.e. with the assumption of white process noise. The fundamental intent in the use of such a Kalman filter design cascaded with the linear regulators of Section 4.2 was to remove high frequency disturbances from the controller. The problem of measurement noise is considered, for the purpose of the present analysis, to be a different issue. Its form does, of course, impact the filter design in no less a manner than process noise, as has been demonstrated by Sugimoto and Kojima [9]. For purposes of this analysis, however, the measurement noise is assumed to be white as has also been assumed by Astrom [13,36].

Based on the seaway amplitude and disturbance models of Section 2 and as shown in Figs. 2 through 4, the sway force and yaw moment spectra relevent to the ship in the full load condition at cruise speed in Beaufort 8 conditions at a 90 degree seaway encounter angle are shown in Figs. 12 and 13, respectively. The RMS values were used as the white noise spectral density approximations to these disturbances for the filter design--0.751 x 10^{14} lb^2 - second and 0.754 x 10^{18} $(lb\text{-}ft)^2$-second, respectively, for sway force and yaw moment. The corresponding values for the ballast case are 0.288 x 10^{14} lb^2-second and 0.162 x 10^{16} $(lb\text{-}ft)^2$-second, respectively. Steady-state Kalman gains were computed on the basis of the continuous model of the ship/steering system of Section 2 (Eqs. (2) and (5)) with the assumption of a heading measurement from the gyrocompass with an RMS measurement error of 0.25 degree.

With the system represented in the form

$$\dot{X}' = A'X' + B'u + Dw \tag{13}$$

where X' is the addended state vector (to include the effect of rudder dynamics) $(v,r,\psi,\delta)^T$, $u = \delta_c$,

$$A' = \left[\begin{array}{c|c} A & B \\ \hline 0 & -1/\tau \end{array}\right] , \tag{14}$$

and

$$B' = \left[\begin{array}{c} 0 \\ \hline 1/\tau \end{array}\right], \tag{15}$$

then the combined estimator/controller equations are

$$\dot{\hat{X}} = A'\hat{X} + C[z - H\hat{X}] + B'u \tag{16}$$

$$u(t) = -K X (t) \tag{17}$$

and

$$z = HX + \nu \tag{18}$$

where: $\hat{X}$ is the estimated state vector $(\hat{v},\hat{r},\hat{\psi},\hat{\delta})^T$,

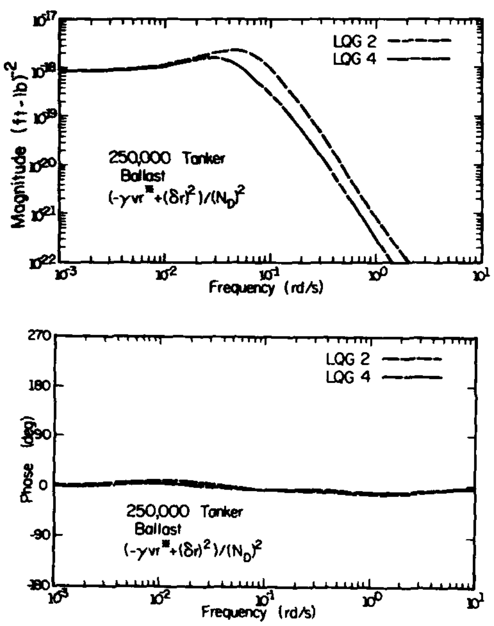

Figure 11 Spectral Density of Total Added Resistance/Spectrum of Yaw Moment: 250,000 DWT Tanker, Ballast, Controllers LWG 2 and LWG 4

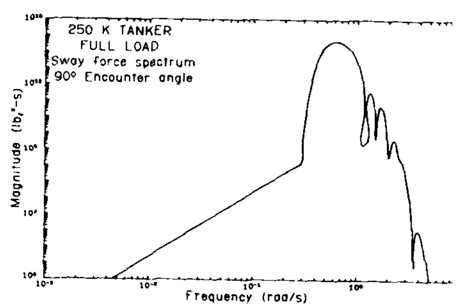

Figure 12 Sway Force Spectrum, 90 Degree Encounter Angle: 250,000 DWT Tanker Full Load, Beaufort 8 Conditions

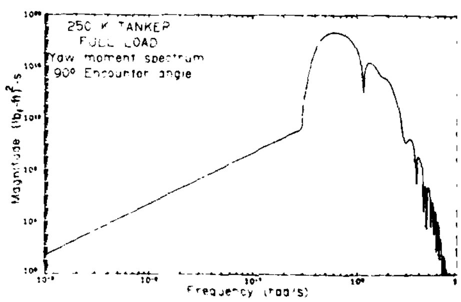

Figure 13 Yaw Moment Spectrum, 90 Degree Encounter Angle: 250,000 DWT Tanker Full Load, Beaufort 8 Conditions

C is the Kalman gain matrix $[C_1, C_2, C_3, C_4]^T$,

z is the gyrocompass measurement, and

H = [0 0 1 0], the measurement matrix, and v is gyrocompass measurement noise. With the assumptions stated above and use of the linear regulator laws of Section 4.2 (Table 4), then, by the separation theorem [37], the resulting controllers will minimize the expected value of Eq. [12] for each of the different values of weighting coefficient λ (Table 4). The steady-state Kalman gains computed from solution of the continuous form of the resulting Riccati equation [37] for the full-load and ballast cases were:

Full Load

$C^T = [-45.108,\ 0.077,\ 0.394,\ 0]$

Ballast

$C^T = [-83.027,\ 0.043,\ 0.292,\ 0]$

Notice, as pointed out by Parsons and Cuong [38] who have used a similar design process for the different problem of path control in shallow water, that the filter does nothing to improve the existing estimate of rudder angle . This is because its response to the commanded rudder angle δ_c known (Eq. 5). This obviates the need for measurement of rudder in any online filter implementation other than for the very important function of failure detection.

4.5 Combined Kalman Filter and Linear Regulator Performance

The estimator/controller equations (Eqs. (16)-(18)) were included in the time-domain simulation to replace the controller equations of Section 4.2 (Table 4) based on the assumption of perfect state knowledge. To obtain a direct comparison of performance between the combined Kalman filter/linear regulator design and the "perfect" linear regulator design, both based on the assumption of white process noise, no measurement noise was included in the gyrocompass measurement of yaw in this particular study. In justification of this, the purpose of the study was to test the validity of the assumption of white process noise in the LQG formulation of the problem, and to measure the ability of the filter to reduce the high frequency disturbances to the controller in the actual seaway environment in any event.

The results of these Kalman filter/linear regulator studies, using the steady-state Kalman gains of Section 4.4 for full-load and ballast conditions both with the higher and lower values of yaw penalty coefficient λ as shown in Table 4, are summarized in Table 6 and Figs. 14 through 17. The following designations are used in definition of the various LQG controllers:

	Weighting Coefficient λ	Kalman filter/LR Controller Designation LQG
Full Load:	29.5	5
	7.4	7
Ballast:	15.8	6
	3.9	8

It is apparent that, as in the case of "perfect" LR studies, the lower bandwidth LQG controllers, LQG7 and LQG8 in general outperform those designed to the higher yaw penalty coefficient, LQG5 and LQG6, in terms of net added resistance (Table 6

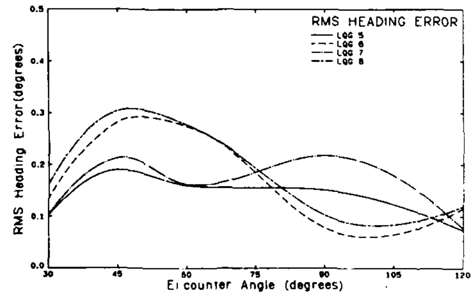

Figure 14 RMS Heading Error: 250,000 DWT Tanker, Beaufort 8, 15 Knots, Controllers LQG 5-LQG 8

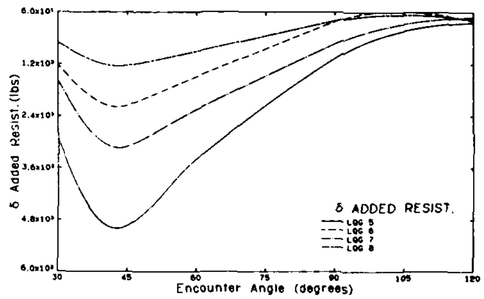

Figure 16 Mean Induced Rudder Drag: 250,000 DWT Tanker, Beaufort 8, 15 Knots, Controllers LQG 5-LQG 8

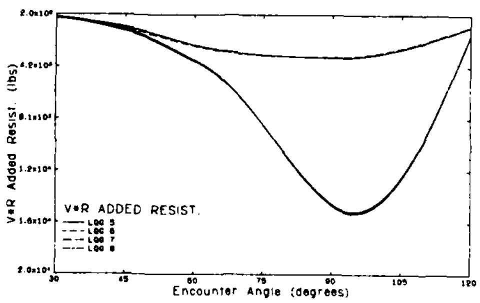

Figure 15 Mean Yaw/Sway Component of Added Resistance: 250,000 DWT Tanker, Beaufort 8, 15 Knots, Controllers LQG 5-LQG 8

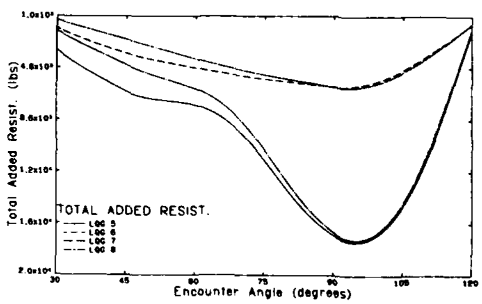

Figure 17 Total Mean Added Resistance due to Steering: 250,000 DWT Tanker, Beaufort 8, 15 Knots, Controllers LQG 5-LQG 8

Table 6 Steering Performance of LQG Controllers 5-8: 250,000 DWT Tanker, 15 Knots, Beaufort 8

Encounter Angle (Deg)	Yaw (Deg)	Mean Added Resistance (lb_f) Yaw/Sway	Sway^2	Rudder	Total
Full Load					
LQG No. 5					
30	0.10	470	2980	40	3490
45	0.19	1440	4950	130	6510
60	0.16	3750	3410	380	7540
90	0.15	14620	1110	1330	17060
120	0.07	1720	300	120	2150
LQG No. 7					
30	0.10	400	1630	30	2070
45	0.21	1330	3120	120	4580
60	0.16	3670	2280	380	6330
90	0.22	14720	780	1330	16840
120	0.08	1740	170	120	2040
Ballast					
LQG No. 6					
30	0.14	440	1280	170	1890
45	0.28	1250	2190	300	3750
60	0.27	2580	1540	700	4820
90	0.08	3420	270	2470	6160
120	0.12	1060	260	270	1590
LQG No. 8					
30	0.16	390	750	170	1310
45	0.30	1150	1280	290	2720
60	0.28	2500	950	700	4150
90	0.10	3460	260	2480	6190
120	0.12	1080	220	270	1580

and Fig. 17). Controllability afforded by all controllers (Fig. 14 and Table 6) is very good. The higher yaw penalty controllers show only slight improvement over those designed to the lower yaw penalty in yaw control. It also seems fairly apparent that no controller is able to substantially affect the basic yaw/sway components of added resistance of the ship (Fig. 15 and Table 6). The issue seems to become, then, in terms of propulsion efficiency almost that of which controller can maintain control of the ship with the minimum amount of rudder, as is demonstrated by Fig. 16, which shows a comparison of rudder induced drag resulting from the various controllers.

Stability is virtually assured from linear regulator-based controllers designed to the performance criterion of Eq. (12) [37]. Due to the inherent instability with the resulting conditional closed-loop stability of the tanker at full-load (Table 2) [39] operator setting of conventional autopilot control parameters is somewhat risky. LQG controller design therefore seems attractive, at first sight, for this type of ship. A comparison, however, of the performance of these combined Kalman filter/linear regulators controllers with that of the "perfect" linear regulator controllers and with that of the conventional autopilot is rather instructive. Examination of Table 5 (LQG controllers 1-4) and Table 6 (LQG controllers 5-8) shows that the addition of the Kalman filter has substantially reduced the losses of the fully-loaded ship in beam seas (e.g. the losses from the "perfect" LR controller LQG3 have been reduced from 21,750 pounds force mean added resistance, corresponding to a propulsion loss of some 4.77 percent of normal full power, to 16,840 pounds force-3.69 percent loss, by the corresponding LQG controller with Kalman filter using only heading measurement LQG7. The overall performance improvements in other conditions are not very substantial, however, and propulsion performance is actually degraded by the addition of the filter in following/quartering seas.

These results demonstrate, as also indicated by the representations of seaway disturbances in Figs. 12 and 13, that the assumption of white noise for the seaway disturbance is not valid in beam seas, at least. There is no doubt, however, that the Kalman filter, albeit designed on the basis of an apparently rather weak assumption, was successful in effecting a substantial reduction in rudder activity in beam seas, and did also bring about a slight reduction in the yaw/sway components of added resistance. To further examine the filter's effect on system performance, Figs. 18 through 20 present plots of the ship states and estimated ship states, resulting from simulation studies of controller LQG7 (full-load, low yaw coefficient penalty combined Kalman filter/linear regulator design) for a 90 degree encounter angle. The reasons for the reduction in RMS rudder angle are apparent. The relatively high frequency motions of the automatically controlled ship result from the relatively high gains necessary to stabilize the inherently unstable hull in the full-load condition. It is worth noting also that with a controller designed to the performance criterion of Eq. (12), sway velocity remains largely unaffected by choice of controller. In contrast to these results, the filter was seen to overestimate the controlled ship states in following/quartering seas.

Examination of the performance of the conventional autopilot (Table 3) shows, however, that any of these LQG controllers are outperformed by controllers CV2 and CV3 almost across the board in terms of propulsion efficiency at full-load, with following/quartering seas presenting, apparently, particular difficulty to the LQG controllers. To determine what, if any, improvement in performance of LQG controllers could be achieved by modeling of the seaway disturbances by exponentially correlated noise rather than white noise [37], a Kalman filter was designed on the basis of the relevant sway force and yaw moment approximations to the 90 degree encounter angle disturbance spectra of Figs. 12 and 13 of for the full-load case. With only a yaw deviation measurement the seaway disturbances are not fully observable, and no attempt was made in this study to include the disturbance states in the control law, as was done, for example, in [38]. Simulation studies were conducted to assess the performance of LQG controllers designed on this basis. The control laws used were identical to those defined in Section 4.2 and used in the other studies. The intent was simply to determine if this modeling of the seaway disturbances in the Kalman filter design would result in better controller performance. The resulting steady-state Kalman gains used in the simulations in determination of the state vector X' (Eq. (16)) based on a heading measurement with the same assumed noise statistics were:

<u>Full Load</u>

$$C^T = [-14.035,\ 0.017,\ 0.182, 0]$$

The reduction in the Kalman gains from those derived on the basis of white process noise approximations (Section 4.4) is apparent.

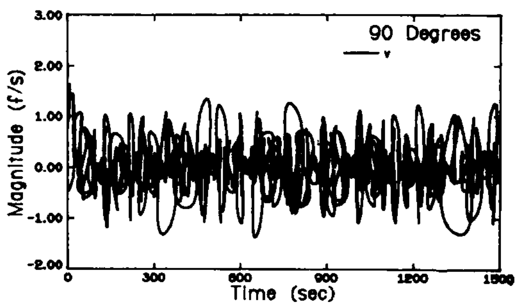

Figure 18a Sway Velocity at 90 Degree Encounter Angle: 250,000 DWT Tanker, Beaufort 8, 15 Knots, Controller LQG 7

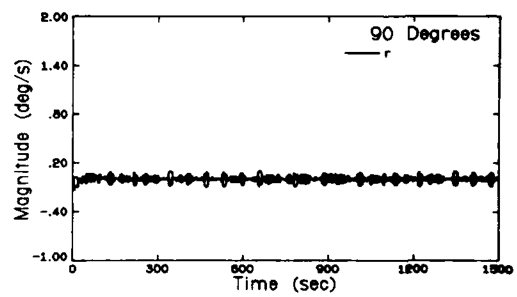

Figure 19a Yaw Rate at 90 Degree Encounter Angle: 250,000 DWT Tanker, Beaufort 8, 15 Knots Controller LQG 7

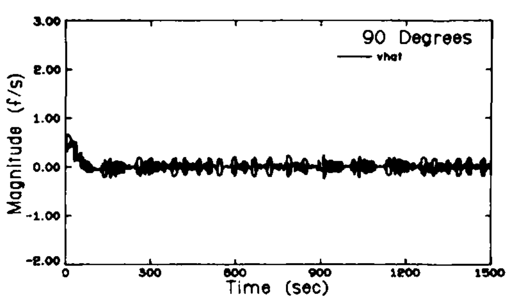

Figure 18b Estimated Sway Velocity at 90 Degree Encounter Angle: 250,000 DWT Tanker, Beaufort 8, 15 Knots, Controller LQG 7

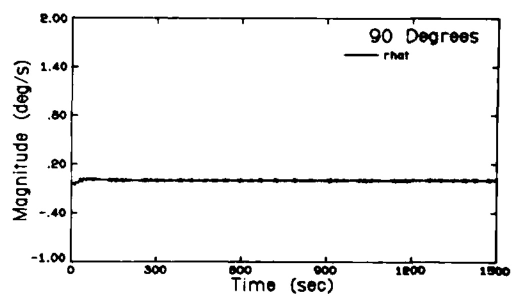

Figure 19b Estimated Yaw Rate at 90 Degree Encounter Angle: 250,000 DWT Tanker, Beaufort 8, 15 Knots, Controller LQG 7

The resulting performance of the combined full-load amended Kalman filter/linear regulator designed to the lower yaw penalty (Table 4), designated LQG11, for the ship in the full-load condition with, as before, no heading measurement noise present is summarized in Table 7. The corresponding full-load LQG controllers with which it can be compared are LQG3 designed on the basis of white noise disturbances and perfect state knowledge, and LQG7 designed on the basis of white noise disturbances and imperfect yaw measurement. The performance of these controllers for the fully-loaded ship is summarized in Tables 5 and 6, respectively. Controller LQG11 shows an across-the-board improvement in added resistance performance over controllers LQG3 and LQG7. Yaw deviation performance is comparable.

The results provide further proof, however, that the yaw/sway components of added resistance, the major cause of propulsion losses in beam seas, remain almost unaffected by choice of controller. The results also provide apparent proof of the superior performance of Kalman filters designed on the basis of exponentially correlated noise rather than white noise as an approximation for the actual seaway disturbances for all seaway directions, and most particularly for following/quartering seas. Comparison of these results with those for the conventional autopilot under the same conditions (Table 3) shows, however, that controller LQG11 cannot, in general, match the overall performance of the conventional autopilot with settings corresponding to CV2 or CV3.

4.6 Fixed-Structure Controller Design and Performance

While the apparent failure of LQG controllers to outperform a conventional autopilot, on the basis of these results, seems clear, the difficulties associated with the latter are acknowledged. Operator setting of control paramters which, in general, have little relation to the proper specification of these for either controllability or propulsion efficiency serves to underline the unsatisfactory nature of such autopilots. This would appear to be especially true of their use on VLCC's. This is not true, however, for designs based on dynamic controllers in a limited state feedback implementation, where the selection of controller parameters may be wholly adaptive [6,7] or by simple operator selection of speed and load [6,7]. Such controllers, designed to a performance criterion similar to Eq. (12) have demonstrated their ability to reduce propulsion losses and improve controllability on high-speed containerships [7].

The design of such a controller for the ship under discussion in this paper is discussed in [39]. To illustrate the apparent validity of a system based on a relatively simple dynamic lead/lag controller, designated LL1, optimized to the performance criterion of Eq. (12) using the higher value of yaw penalty coefficients in both full-load and ballast cases (Table 4) [39], the results of two simulation studies for the fully-loaded ship are presented. The first study compares the time-domain results of the ship controlled by the dynamic lead/lag controller with those for the ship controlled by the best combined Kalman filter/linear regulator, LQG11, with no yaw measurement noise present. The results for both controllers appear in Table 7. The lead/lag controller is seen to outperform the LQG controller in terms of propulsion efficiency at all but the 120 degree encounter angle. Yaw deviation resulting from the use of either controller is fairly comparable except at 120 degrees encounter angle, where LQG11 shows better controllability and also uses less rudder. Otherwise, controller LL1 uses the lesser rudder which accounts for its superior performance. It is clear, from inspection of these results that sway velocity, yaw rate, and the resulting yaw/sway components of added resistance are unaffected by the choice of controller. The issue very clearly for this particular type of ship seems to depend on controller bandwidth and cut-off characteristics. In this respect, it is worth noting that the conventional

Table 7 Steering Performance of Controllers LQG 11 and LL1
250,000 DWT Tanker: Full Load, 15 Knots, Beaufort 8

Encounter Angle (Deg)	Sway Velocity (ft/s)	Yaw Rate (Deg/s)	Yaw (deg)	Rudder Angle (Deg)	Rudder Rate (Deg/s)	Mean Added Resistance (lb_f)			
						Yaw/Sway	$Sway^2$	Rudder	Total
LQG 11									
30	0.09	0.03	0.10	0.90	0.25	250	30	280	560
45	0.16	0.05	0.23	1.72	0.35	1090	120	1040	2240
60	0.29	0.07	0.16	0.94	0.40	3290	360	300	3350
90	0.55	0.04	0.23	1.34	0.16	14810	1340	640	16800
120	0.17	0.05	0.08	0.39	0.12	1790	120	50	1970
LL1									
30	0.09	0.03	0.20	0.43	0.11	360	30	60	450
45	0.16	0.05	0.18	0.51	0.18	990	110	90	1180
60	0.28	0.07	0.17	0.52	0.22	3280	360	100	3730
90	0.55	0.04	0.43	0.58	0.12	14830	1330	120	16280
120	0.18	0.05	0.50	0.68	0.11	2120	140	160	2420

autopilot with its higher cut-off rate characteristics (Eq. (9)), slightly outperforms the simple lead/lag controller, with the CV2 setting (Table 3).

The Kalman filter gains used in the simulation studies of the LQG controllers were based on the assumption of Gaussian white heading measurement noise of 0.25 degree RMS. It might therefore appear that the mismatch caused by the absence of measurement noise in the simulations biased the results in favor of the controllers designed on a more deterministic basis, or those stochastic controllers designed and evaluated with the assumption of perfect state knowledge. A simulation study was therefore conducted to determine the effect of heading measurement noise of the above statistics on controllers LQG11 and LL1 for the fully-loaded ship. The results of this study are shown in Table 8. From these it is clear that, though both controllers show degraded performance as compared to the results obtained with the presence of no measurement noise (Table 7), the lead/lag controller show superior ability to filter the measurement noise than does the Kalman filter. Figure 21 shows the yaw deviation measurement at 90 degree encounter angle for the LQG11 controlled ship at full-load. Figures 22 and 23 show, respectively, the Kalman filter's estimate of yaw and the yaw performance of the controlled ship. The yaw deviation measurement and actual yaw deviation resulting from controller LL1 are shown in Figs. 24 and 25. The effective bandwidth of the controllers is therefore a fundamental issue in attenuation of both process and measurement noise. In this respect, it appears that a properly designed limited state feedback controller with dynamics in the feedback can be expected to perform better than LQG controllers based on Kalman filter designs such as have been proposed in [12,13].

It has also been found by Kallstrom, et al. [12], that the addition of a Kalman filter to a conventional PID controller improved system performance. To determine the effects of such a scheme, the Kalman filter based on exponentially correlated process noise was cascaded with the lead/lag controller LL1, and the resulting performance evaluated for the fully-loaded ship in the presence of the same measurement noise as simulated above. The steady-state performance of this estimator/controller combination, designated KF/LL1 is summarized in Table 8. While the results show that only marginal improvement in overall performance over the simple lead/lag controller is achieved, they appear to offer further proof that the LQG controllers presented are not optimal in minimization of added resistance in the actual seaway environment.

5. *Implications for Autopilot Design*

On the basis of the results presented in this paper, it appears that steering controllers based on limited state-feedback with dynamics in the feedback loop offer a viable alternative to conventional ship autopilots and LQG controllers based on simplified disturbance noise assumptions [12,13] and designed to the form of performance criterion of Eq. (12), for control of a large tanker in a seaway. As such, the results raise significant questions as to the validity or usefulness of such a stochastic approach to controller design in the environment afforded by a relatively large, slow speed ship in a seaway.

For further improvements in added resistance performance using LQG techniques, it would appear that an alternative form of criterion based on Eq. (4) [25,27] should be tried as a basis for controller design. With controllers designed to the criterion of Eq. (12), neither yaw rate nor sway velocity are penalized, and the resulting added resistance components due to these effects seem unaffected by controller design. It also seems that a more adequate representation of seaway disturbances is required in the model used for the state estimator than the simplified models which have been commonly used. The work by Sugimoto and Kojimio [9] and Grimble, et al. [40], is indicative of the kind of modeling of the seaway that may be necessary, but also may present fairly severe problems in any attempted online implementation [28].

Table 8 Effect of Measurement Noise on Steering Performance of Controllers LQG 11 and LL1

Encounter Angle (Deg)	Sway Velocity (ft/s)	Yaw Rate (Deg/s)	Yaw (deg)	Rudder Angle (Deg)	Rudder Rate (Deg/s)	Mean Added Resistance (lb_f)			
						Yaw/Sway	$Sway^2$	Rudder	Total
LQG 11 with measurement noise									
30	0.09	0.03	0.13	2.14	0.55	330	30	1560	1920
45	0.16	0.05	0.23	2.36	0.59	1140	120	1940	3200
60	0.29	0.07	0.17	1.97	0.61	3320	360	1330	5020
90	0.55	0.04	0.25	2.53	0.54	14860	1340	2220	18410
120	0.17	0.05	0.11	2.01	0.51	1850	120	1380	3350
LL1 with measurement noise									
30	0.08	0.03	0.20	0.87	0.91	260	30	260	550
60	0.29	0.07	0.21	0.95	0.89	3360	360	310	4040
90	0.56	0.04	0.75	1.32	0.92	15300	1370	620	17290
120	0.17	0.05	0.19	0.81	0.89	1810	120	230	2160
KF/LL1 with measurement noise									
30	0.08	0.03	0.19	0.49	0.06	220	30	80	330
45	0.17	0.05	0.55	0.87	0.06	1200	120	270	1590
60	0.28	0.07	0.20	0.41	0.05	3220	360	60	3640
90	0.56	0.04	0.88	1.19	0.06	15500	1390	500	17390
120	0.17	0.05	0.14	0.42	0.05	1820	120	60	2000

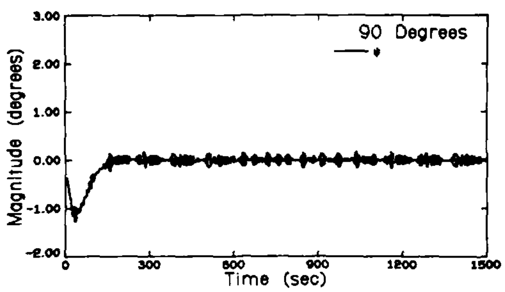

Figure 20a Yaw at 90 Degree Encounter Angle: 250,000 DWT Tanker, Beaufort 8, 15 Knots, Controller LQG 7

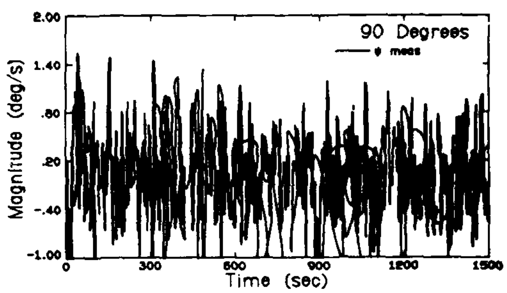

Figure 21 Yaw Measurement at 90 Degree Encounter Angle: 250,000 DWT Tanker, Beaufort 8, 15 Knots, Controller LQG 11 with Measurement Noise

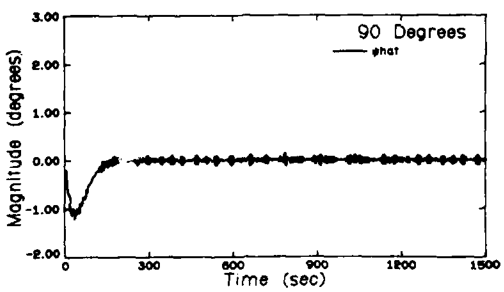

Figure 20b Estimated Yaw at 90 Degree Encounter Angle: 250,000 DWT Tanker, Beaufort 8, 15 Knots, Controller LQG 7

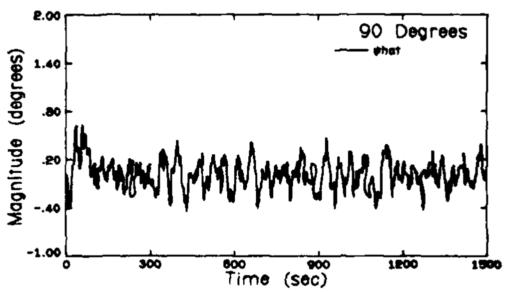

Figure 22 Estimated Yaw at 90 Degree Encounter Angle: 250,000 DWT Tanker, Beaufort 8, 15 knots, Controller LQG 11 with Measurement Noise

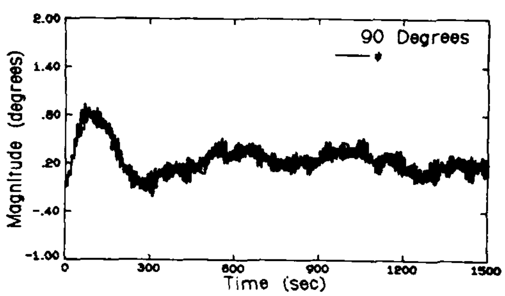

Figure 23 Yaw at 90 Degree Encounter Angle: 250,000 DWT Tanker, Beaufort 8, 15 Knots, Controller LQG 11 with Measurement Noise

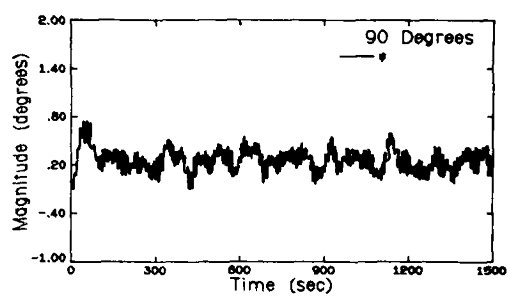

Figure 25 Yaw at 90 Degree Encounter Angle: 250,000 DWT Tanker, Beaufort 8, 15 Knots, Controller LL1 with Measurement Noise

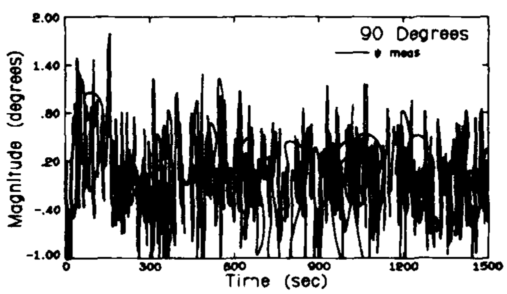

Figure 24 Yaw Measurement at 90 Degree Encounter Angle: 250,000 DWT Tanker, Beaufort 8, 15 Knots, Controller LL1 with Measurement Noise

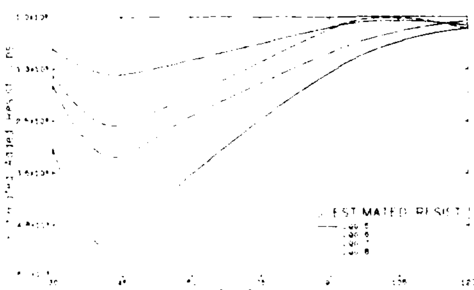

Figure 26 "Approximate" Mean Total Added Resistance due to Steering as Estimated Online: 250,000 DWT Tanker, Beaufort 8, 15 Knots, Controllers LQG 5-LQG 8

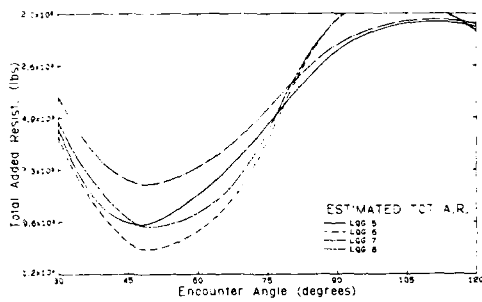

Figure 27 Online Estimate of Mean Total Added Resistance due to Steering: 250,000 DWT Tanker, Beaufort 8, 15 Knots, Controllers LQG 5-LQG 8

The results presented are significantly different from those obtained from examination of the steering performance of a containership [25]. They suggest that the basic dynamics of the VLCC steering problem, which include a relatively low rudder area-to-hull cross sectional area ratio, large ship inertia, and relatively low ship speed resulting in relatively high wave encounter frequencies, may well work against any further attempt to reduce the hull inertia added resistance in a seaway by improved controller design. The controller design problem in that case simplifies to that of minimum bandwidth design to meet certain performance requirements [41], which in the case of a VLCC at full-load, include stability. The results presented here and in [26,39] give further weight to this proposition. They also indicate that no adaptivity to seaway encounter angle seem necessary for the case of the control of a large tanker. This is not the case for high-speed ships such as, for example, containerships [7,25,42]. Given the increasingly well defined data base for the hydrodynamic characteristics of such ships [14] the use of a relatively simple steering controller implementation based on "a priori" optimization of classical PID controller parameters at design conditions seems well justified for the particular type of hull examined. It seems particularly applicable to tankers since, typically, there are only two loading conditions and one operational speed during open-seas course-keeping.

Several recent steering controller designs have the potential to provide online identification of system dynamics with reduced knowledge of either ship or seaway parameters [9,11-13]. Any attempt, however, to minimize added resistance by online minimization of the form of performance criterion of Eq. (12) in an adaptive controller scheme would seem to present serious difficulties in the case of the ship examined. Figure 26 shows the relationship between the approximate criterion J (Eq. (14)) scaled to pounds force as estimated online by the LQG controllers 5 and 7 (full-load) and 6 and 8 (ballast) i.e. the linear regulators cascaded with Kalman filters based on white noise, and encounter angle for ship speed of 15 knots in Beaufort 8 conditions. The curves bear little resemblance to those already presented in Fig. 17 for the same cases showing the actual mean added resistance based on Eq. (4). Online minimization of an index based on the expected value of Eq. (4) with the system states estimated on the basis of a white noise disturbance Kalman filter design does not appear to have the potential to minimize the actual added resistance either, as may be seen from the curves of $\overline{\Delta X}$ of Fig. 27. These results give further support to the use of a relatively simple controller based on "a priori" optimization for this type of ship.

ACKNOWLEDGEMENTS

This work has been supported by the United States Maritime Administration under the University Research Program and by the Research Board of the University of Illinois at Urbana-Champaign. The authors thank Dr. H. Eda at the Davidson Laboratory of Stevens Institute of Technology, Hoboken, New Jersey, for his help in the definition of hydrodynamic data for the ship.

REFERENCES

1. Highfill, J. H., III, and W. T. Spurgin, "Improvement in Steering Control for Merchant Ships," Nat. Maritime Res. Center Report NMRC-KP-138, 1975.

2. Highfill, J. H., III, and W. T. Spurgin, "Improvement in Steering Control for Merchant Ships," Nat. Maritime Res. Center Report NMRC-KP-138A, 1976.

3. Reid, R. E., "Improvement in Steering Control for Merchant Ships, Phase IIA," Final Report, Nat. Maritime Res. Center Report NMRC-188, 1978.

4. Reid, R. E., and V. E. Williams, "A New Ship Control Design Criterion for Improving Heavy Weather Steering," Proc., 5th Ship Control Systems Symp., Annapolis, Md., 1978.

5. Reid, R. E., "Design of an Automatic Steering Controller for Ships to Minimize Added Resistance due to Steering," Ph.D. Dissertation, University of Virginia, April 1978.

6. Reid, R. E., and J. W. Moore, "Optimal Steering Control of High-Speed Containerships: Part I System Dynamics," Proc., 1980 JACC, Aug. 1980.

7. Reid, R. E., and J. W. Moore, "Optimal Steering Control of High-Speed Containerships: Part II Controller Design," Proc., 1980 JACC, Aug. 1980.

8. Tosi, F., and E. Verde, "Microprocessor Based Adaptive Autopilot: System Development and Sea Trials," Proc., Symp. on Ship Steering Automatic Control, Genoa, Italy, June 1980.

9. Sugimoto, A., and T. Kojima, "A New Autopilot System with Condition Adaptivity," Proc., 5th Ship Control Systems Symp., Annapolis, Md., 1978.

10. Schilling, A. C., "Economics of Autopilot Steering Using an IBM System 7 Computer," Proc., 2nd IFAC/EFIP Symp. on Ship Operation Automation, Washington, D.C., 1976.

11. van Amerongen, J., and H. R. van Nauta Lemke, "Optimum Steering of Ships with an Adaptive Autopilot," Proc., 5th Ship Control Systems Symp., Annapolis, Md., 1978.

12. Kallstrom, C. G., et al., "Adaptive Autopilots for Tankers," Automatica, Vol. 15, 1979.

13. Astrom, K. J., "Design of Fixed Gain and Adaptive Autopilots based on the Nomoto Model," Proc., Symp. on Ship Steering Automatic Control, Genoa, Italy, June 1980.

14. Eda, H., R. Falls, and D. A. Walden, "Ship Maneuvering Safety Studies," Trans., SNAME, Vol. 87, 1979.

15. Reid, R. E., M. Youhanaie, and A. K. Tugcu, "Describing Function Analysis of Ship Steering System Nonlinearities," Proc., 1981 JACC, Charlottesville, Va., June 1981.

16. Tomita, Y., H. Yamamoto, K. Okita, and K. Yamazaki, "The Single Loop Steering System [Ships]," Hitahi Zosen Tech. Rev. (Japan), Vol. 41, No. 2, June 1980, 10-15.

17. Eda, H., and C. L. Crane, Jr., "Steering Characteristics of Ships in Calm Water and Waves," Trans., SNAME, Vol. 73, 1965.

18. Abkowitz, M. A., "Lectures on Ship Hydrodynamics-Steering and Maneuverability," Hy A-Report No. Hy-5, Lygnby, Denmark, 1964.

19. Norrbin, N. H., "Theory and Observations on the Use of a Mathematical Model for Ship Maneuvering in Deep and Confined Waters," Proc., 8th Symp. on Naval Hydrodynamics, Pasadena, Cal.,Aug. 1970.

20. Norrbin, N. H., "On the Added Resistance due to Steering on a Straight Course," 13th ITTC Report of Peformance Committee, 1972.

21. Reid, R. E., "Performance Criteria for Propulsion Losses due to Ship Steering," J. of Dynamic Systems, Measurement, and Control, to be published.

22. Bech, M. I., "Some Aspects of the Stability of Automatic Course Control of Ships," J. of Mech. Engr. Sci., Vol. 14, No. 7, 1972.

23. Gie, T. S., "Model Tests on a 900-foot Containership," NSMB (The Netherlands) Report No. 69-122-2T, Nov. 1969.

24. Michel, W. H., "Sea Spectra Simplified," Marine Tech., Jan. 1968.

25. Reid, R. E., and P. F. Parent, "The Application of Linear Regulator Techniques to Minimization of Steering Losses Suffered by a Ship in a Seaway," Proc. 1981 JACC, Charlottesville, Va., June 1981.

26. Reid, R. E., and B. C. Mears, "Design of the Steering Controller of a Supertanker using Linear Quadratic Control Theory: A Feasibility Study," Proc., 20th CDC, San Diego, Cal., Dec. 1981.

27. Reid, R. E., B. C. Mears, and D. E. Griffin, "Energy Loses Related to Automatic Steering of Ships: Their Evaluation and Minimization," ASME Ocean Engr. Div., 1981 Winter Annual Meeting, Washington,D.C., Nov. 1981.

28. Parent, P. F., "Application of Modern Control Theory to the Steering System of a Ship," M.S. Thesis, Dept. of Mech. and Ind. Engr., Univeristy of Ill. at U-C, July 1980.

29. Nomoto, K., and T. Motoyama, "Loss of Propulsive Power Caused by Yawing with Particular Reference to Automatic Steering," Jap. Shipbuilding and Marine Engr., Vol. 120, 1966.

30. Motora, S., and T. Koyama, "Some Aspects of Automatic Steering of Ships," Jap. Shipbuilding and Marine Engr., Vol. 3, No. 4, July 1968.

31. Blanke, M., "On Identification of Nonlinear Speed Equation from Full Scale Trials," Proc., 5th Ship Control Systems Symp., Annapolis, Md., Oct. 1978.

32. Ohtsu, K., M. Horigome, and G. Kitagwa, "A New Ship's Autopilot Design through a Stochastic Model," Automatica, Vol. 15, 1979.

33. van Amerongen, J., and H. R. van Nauta Lemke, "Criteria for Optimum Steering of Ships," Proc., Symp. on Ship Steering Automatic Control, Genoa, Italy, June 1980.

34. Kallstrom, C. G., and N. H. Norrbin, "Performance Criteria for Ship Autopilots: An Analysis of Shipboard Experiments," Proc., Symp. on Ship Steering Automatic Control, Genoa, Italy, June 1980.

35. Clarke, D., "Development of a Cost Function for Autopilot Optimization," Proc., Symp. on Ship Steering Automatic Control, Genoa, Italy, June 1980.

36. Astrom, K. H., "Why use adaptive techniques for steering large tankers?," Int. J. Control, 1980, Vol. 32, No. 4, 689-708.

37. Bryson, A. E., Jr., and Y. C. Ho, Applied Optimal Control, revised printing, Wiley and Sons, 1975.

38. Parsons, M. G., and H. T. Cuong, "Optimal Stochastic Path Control of Surface Ships in Shallow Water," Office of Naval Res. Report ONR-CR215-249-2 F, Aug. 1977.

39. Reid, R. E., and D. E. Griffin, "Steering Dynamics of a Very Large Crude Carrier," submitted to J. Dynamic Sytems, Measurement, and Control.

40. Grimble, M. J., et al., "Use of Kalman Filtering Techniques in Dynamic Ship-Positioning Systems," IEE Proc., Vol. 127, Pt. D, No. 3, May 1980.

41. Newton, G. C., Jr., L. A. Gould, and J. F. Kaiser, Analytical Design of Linear Feedback Controls, Wiley and Sons, ,1957.

42. Reid, R. E., "Automatic Steering Performance of a High-Speed Containership in a Seaway," submitted to J. of Dynamic Systems, Measurement, and Control.

THE USE OF SIMULATION IN THE ANALYSIS OF SHIP STEERING CHARACTERISTICS USING COMBINED ANALOGUE AND DIGITAL TECHNIQUES INVOLVING AUTO-PILOT, SHIP AND ENVIRONMENTAL DISTURBANCE

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ABSTRACT

The purpose of the paper is two-fold, but is mainly aimed at presenting the results of a laboratory analysis into the steering characteristics exhibited by three different vessels under autopilot control resulting from changing autopilot parameters and sea state conditions.

Essentially the paper is in two parts. The first deals with the design of the mathematical models of an autopilot and a ship. Both models have been realised in hardware, as portable desk top units suited for demonstration and teaching, and are coupled together. Both have built in variable parameters which can be adjusted to change the dynamic behaviour of the ship and the dynamic response of the autopilot. A random input of sea spectrum signals has been made available to simulate changes in environmental conditions.

Secondly, the paper will examine the relationship between the autopilot settings and ship behaviour by analogue and digital simulation. Results are correlated and compared. The shortcomings and inadequacy of the models will be discussed.

1. MATHEMATICAL MODELLING OF SHIP AND AUTOPILOT AS PORTABLE DESK TOP UNITS

1.1 The Ship - Mathematics

The availability of a mathematical model to describe the accurate behaviour of a ship is important for design purposes, for laboratory simulation and for testing of ancilliary equipment. The dynamic characteristics of a vessel may be described from a set of equations which are complex and cannot be readily used in basic form for control engineering. Essentially the ship is described as a system having only two inputs and two outputs, i.e. thrust power and rudder angle, and, course and speed respectively.

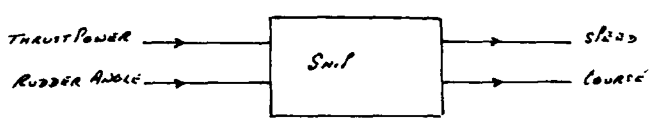


Figure 1. Relating Ship Input and Output.

The derivation of a ship's manoeuvring characteristics can be found from literature by Abkowitz (1), Comstock (2), Nomoto (3), Bech (4) et al but it was Nomoto (5) who derived a model relating the yaw angle as a function of rudder angle and time. It is expressed as a second order differential equation as:-

$$T_1T_2\dddot{\psi} + (T_1 + T_2)\ddot{\psi} + \dot{\psi} = K(T_3\dot{\delta} + \delta) \qquad \text{.....Eqn. 1}$$

where δ = Rudder Angle $\quad\dot{\delta}$ = Rudder Rate
ψ = Yaw Angle $\quad\dot{\psi}$ = Yaw Rate

K, T_1, T_2 and T_3 are the parameters of the hydrodynamic characteristics of the ship.

Expressed in the Laplace domain equation 1 can be written to relate yaw rate to rudder angle as:-

$$\frac{\dot{\psi}}{\delta} = \frac{K(1 + s\,T_3)}{(1 + s\,T_1)(1 + s\,T_2)} \qquad \text{.....Eqn. 2 (Yaw Rate Equation)}$$

Similarly, in order to relate the yaw angle to the rudder angle the equation may be written as:-

$$\frac{\psi}{\delta} = \frac{K(1 + s\,T_3)}{S(1 + s\,T_1)(1 + s\,T_2)} \qquad \text{.....Eqn. 3 (Yaw Angle Equation)}$$

The equations 2 and 3 are linear and assume (i) The ship is stable, (ii) The speed is constant during manoeuvres, and (iii) Rudder angle is limited to $\pm$ 5°.

Bech (4) modified Nomoto's equation to give:-

$$T_1\,T_2\dddot{\psi} + (T_1 + T_2)\ddot{\psi} + KH(\dot{\psi}) = K(T_3\dot{\delta} + \delta) \qquad \text{.....Eqn. 4}$$

which lumped the main non-linearities into a single steering characteristic $H(\dot{\psi}) = \delta$ which by definition is the rudder angle required to outbalance those forces and moments acting upon the hull in a steady turn with rate $\dot{\psi}$.

In general $H(\dot{\psi})$ can be found by the reversed spiral test, or in cases of dynamic stability by the Dieudonne spiral test. Abnormalities in steering characteristics are reflected in abrupt and seemingly random variations in the steering performance. An analysis, within the so called Dieudonne Loop, may look as in figure 2.

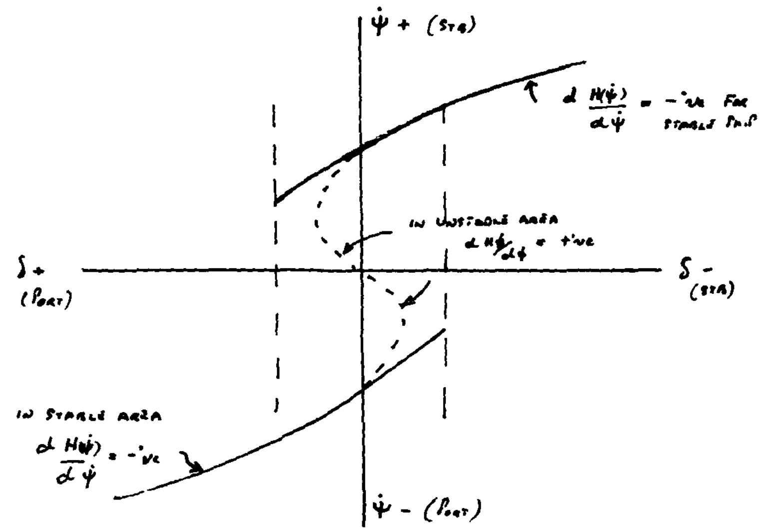


Figure 2. Instability in the Dieudonne Loop

Equation 4 expressed as a transfer function will be:-

$$\frac{\psi}{\delta} = \frac{K(1 + T_3\,S)}{S[T_1\,T_2\,S^2 + (T_1 + T_2)S + K.\,\frac{dH\dot{\psi}}{d\dot{\psi}}]} \qquad \text{.....Eqn. 5}$$

Notice that $\frac{d\,(H\dot{\psi})}{d\dot{\psi}}$ will vary as a function of $\dot{\psi}$, and the ship will be increasing stable as $\dot{\psi}$ increases.

Referring back to the linear Equation 2, relating rate of change of heading to rudder angle for a stable ship, it is possible to see the significance of the terms K, T_1, T_2 and T_3 more clearly. T_1 and T_2 are time constants responsible for a time lag in response to rudder demand. T_3 is a lead term, created by the steady speed of the vessel. K is the static gain. ($\frac{1}{S}$ is an integrating term which only appears on the right hand side of the equation to relate yaw angle to rudder angle as in equation 3.)

By ignoring T_3 in equation 2 we get:-

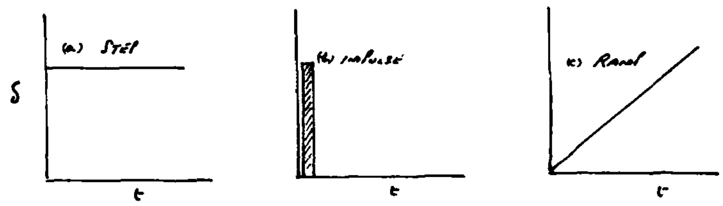
$$\frac{\dot{\psi}}{\delta} = \frac{K}{(1 + T_1S)(1 + T_2S)} \qquad \text{.....Eqn. 6}$$

This is a 2nd order system with a general solution in the time domain of:-

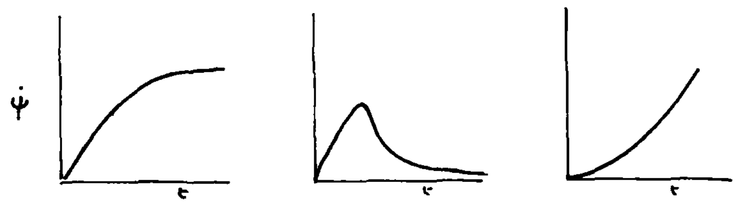
$$\dot{\psi}(t) = C_1 e^{-\frac{t}{T_1}} + C_2 e^{-\frac{t}{T_2}}$$

where C_1 and C_2 are integral constants depending upon initial motion. Provided $\frac{1}{T_1}$ and $\frac{1}{T_2}$ are real and negative, the yaw rate decays with time following a unit disturbance and the yaw angle will again be constant. The ship is stable in this case. (Also, by integrating with the introduction of $\frac{1}{S}$ to equation 6 we obtain yaw angle as before.)

The responses of yaw and yaw angle are shown in figure 3 for a step, an impulse, and a ramp input signal.



(a) Typical Input Signals (δ)



(b) Yaw Rate Output for T.F. $\frac{K}{(1 + sT_1)(1 + sT_2)}$

(c) Yaw Angle Output for T.F. $\dfrac{K}{S[(1 + sT_1)(1 + sT_2)]}$

Figure 3. Responses of Ship Model for 2nd Order System.

In the case of non-linearity and unstable ships, ignoring T_3 in the numerator again, then equation 5 becomes:-

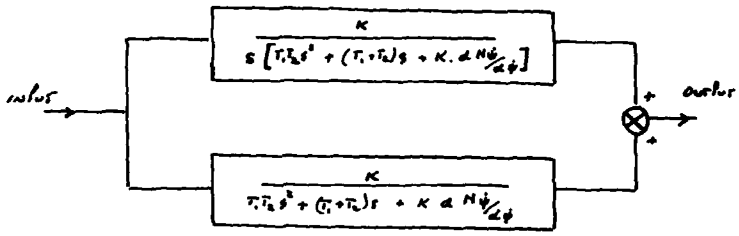
$$\frac{\dot{\psi}}{\delta} = \frac{K}{[T_1 T_2 \, S^2 + (T_1 + T_2)S + K.\, \frac{dH(\dot{\psi})}{d(\dot{\psi})}]}$$

and the condition necessary for $\frac{1}{T_1}, \frac{1}{T_2}$ to be real and negative is such that $\frac{(T_1 + T_2)^2}{4\ K\ T_1\ T_2} \geqslant \frac{dH(\dot{\psi})}{d(\dot{\psi})}$. Hence $\frac{dH(\dot{\psi})}{d\dot{\psi}}$ is the stability index and is determined from the steering characteristic.

By splitting equation 5 into two parts so that:-

$$\frac{\psi}{\delta} = \frac{K}{S[T_1 T_2 \, S^2 + (T_1 + T_2)S + K.\frac{dH(\dot{\psi})}{d\dot{\psi}}]} + \frac{\cancel{S}\ K\ T_3}{\cancel{S}[T_1 T_2 \, S^2 + (T_1 + T_2)S + K.\frac{dH(\dot{\psi})}{d\dot{\psi}}]}$$

the S term in the numerator and denominator cancels so that the response may be considered as a sum of two systems.



In summary the significance of the parameters are:-

(i) K - Static Gain, (ii) $T_1 T_2$ - Time constants for time lag, (iii) $\frac{dH(\dot{\psi})}{d\dot{\psi}}$ - Slope of Steering Characteristic, (iv) T_3 - Lead time due to steering speed of vessel.

1.1.1 The Ship Transfer Function Synthesis

Consider the following transfer function:-

$$\frac{v_0}{v_i} = \frac{K(b_0 + b_1 S)}{a_0 S + a_1 S^2 + a_2 S^3} \qquad \text{.....Eqn. 7}$$

and let $$\frac{v_0^{\,1}}{v_i} = \frac{1}{a_0 S + a_1 S^2 + a_2 S^3} \qquad \text{.....Eqn. 8}$$

then $$\frac{v_0}{v_i} = \frac{v_0^{\,1}}{v_i}\,[K(b_0 + b_1 S)]$$

and $$v_o = K(b_0\, v_0^{\,1} + b_1 S\, v_0^{\,1}) \qquad \text{.....Eqn. 9}$$

Hence, by solving $$\frac{v_0^{\,1}}{v_i} = \frac{1}{a_0 S + a_1 S^2 + a_2 S^3}$$

v_0 will become readily available.

From equation 8,

$$v_i = v_0^{1} \; (a_0S + a_1S_2 + a_2S^3)$$

and, taking the highest derivitive

$$v_0^{1} \; a_2S^3 = v_i - a_0S \; v_0^{1} - a_1S^2 \; v_0^{1}$$

which is represented in the flow diagram illustrated below.

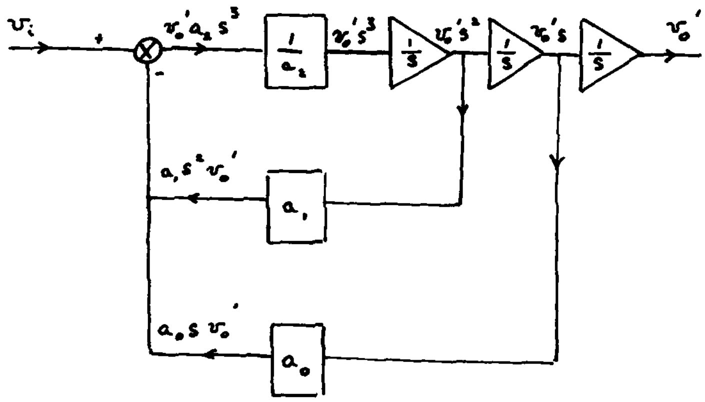


Figure 4. Flow Diagram with Integrators to Solve $\dfrac{1}{a_0S + a_1S^2 + a_2S^3}$

$v_0{}^1$ and S $v_0{}^1$ can be picked off and amplified by K, b_0 and b_1 as in equation 9 in order to give v_0.

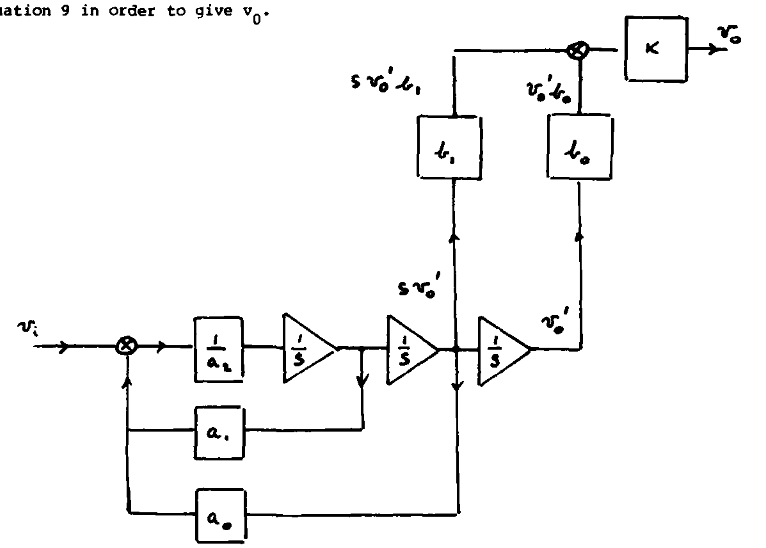


Figure 5. T.F. Synthesis to give $\frac{v_0}{v_i} = \frac{K(b_0 + b_1 S)}{a_0 S + a_1 S^2 + a_2 S^3}$

$$\equiv \frac{\psi}{\delta} = \frac{K(1 + T_3 S)}{K.\frac{dH(\dot{\psi})}{d(\dot{\psi})} S + (T_1 + T_2)S^2 + T_1 T_2 S^3} \quad \text{.....Eqn. 5}$$

By comparing with equation 5 relating ψ to δ, Notice that $b_0 = 1$ $b_1 = T_3$

$a_1 = (T_1 + T_2)$ $\quad a_2 = T_1 T_2$ $\quad a_0 = \frac{K.d.H(\dot{\psi})}{d\dot{\psi}}$

and $a_0 = 1$ for stable and linear ships.

Practical problems in this initial design related to (i) offset, (ii) drift, (iii) practical difficulties in setting parameters precisely, (iv) too many stages resulting in increased problems of drift and offset.

As a result the series of 11 op. amp circuits involved were avoided and passive networks were introduced to perform the same transfer function using integrated circuits which meant the exclusion of the unstable condition. The completed circuit for the linear ship was built, and the parameters were set in at three switches for the time constants T_1, T_2 and T_3, with an adjustment of capacitor size. Forward gain, K, was adjusted between two stages with a variable potentiometer. Yaw angle, and yaw rate were taken from two meters and rudder angle fed in as a voltage derived from the autopilot demand signal. The front panel is shown in figure 6.

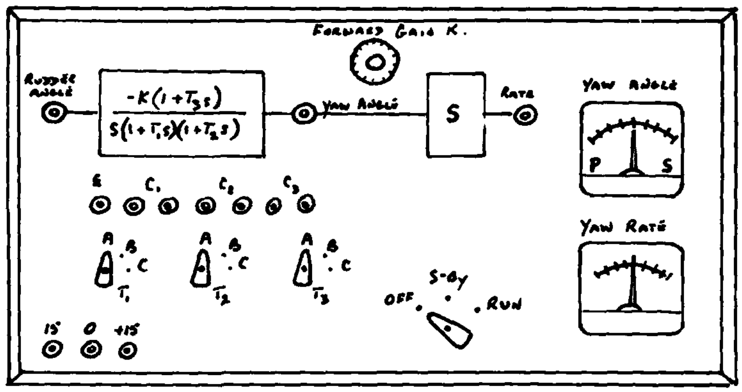


Figure 6. Front Panel of Ship Simulator.

The range of values used were for three linear ship models, namely:-

A	Mariner class vessel (4)	K = 0.052	T_1 = 100	T_2 = 14.4	T_3 = 25
B	M.V. Atlantic Song (6)	K = 0.115	T_1 = 172.3	T_2 = 22.9	T_3 = 95.5
C	M.T. Sea Splendour (6)	K = 0.040	T_1 = 155.6	T_2 = 21.5	T_3 = 31.9

These relate to the settings A, B, C but hybrid values can be obtained by an interchange of switching position for any computation within the range of 27 with any K values up to 10 (corresponding to 0.64 - 23.2) for each of the 27 selections.

1.2 The Automatic Pilot

The front panel of the simulator is shown in figure 7.

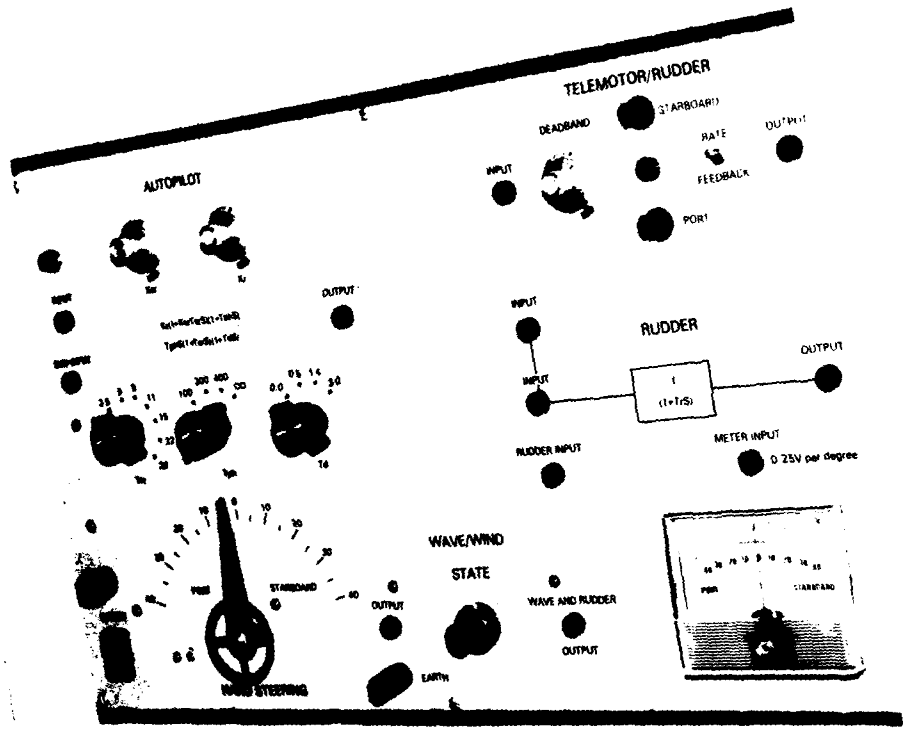
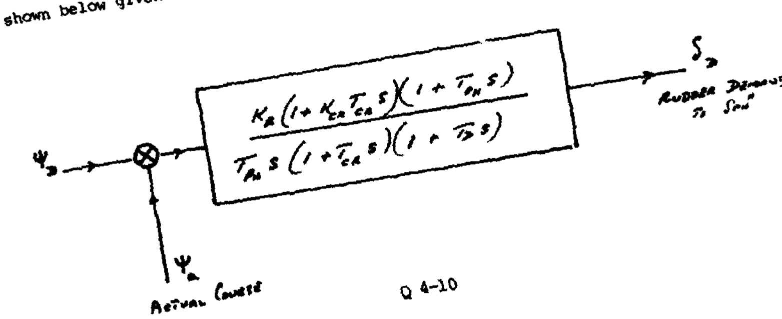


Figure 7. Autopilot Simulator.

The simulator consists of six IC stages and relates to the transfer function shown below given by Bech (4).



where K_R = Rudder Gain

K_{CR} = Counter Rudder Gain

T_{CR} = Counter Rudder Time Constant

T_{PH} = Integral (permanent helm) Time Constant

T_d = Damping Time Constant

The output is a function of the settings of the 5 parameters K_R, K_{CR}, T_{CR}, T_{PH} and T_D. The circuit used contains the standard selectable parameters of proportional, derivative and integral control common to most autopilots and the simulator was based on the Decca DP 750 steering computer.

In fact the simulator contains four separate systems,

(i) the autopilot based on the transfer function marked on the front panel with variable parameters.

(ii) Telemotor and Rudder simulation based on a closed loop mechanical servomechanism using relay control techniques. Deadband width and limit cycles are inherently associated with relay systems and adjustable dead band width was included. Rate feedback was added as an alternative demonstration feature.

(iii) Hand steering which can be used in place of the autopilot, and,

(iv) An external disturbance simulator to provide variable sea/wind signals to the ship simulator.

Each unit can be used as a "stand alone" device or combined (with the ship simulator) to form a complete steering loop. Measuring and recording equipment is attached at output plug positions, and a built in voltmeter (calibrated in both volts and degrees) can be used for connection to any part of the simulator as a general purpose meter.

The control functions on the autopilot are as follows:

(a) Rudder Gain, (K_R), which is a variable potentiometer calibrated from 0.3 to 3.3 in non linear intervals. This determines the absolute degree of rudder command for every degree of steady state heading error. (E.g. with gain control set to 1.0 the rudder moves 1 degree for one degree of heading error.) Used to comply with loading and environmental changes.

(b) Counter Rudder Gain, (K_{CR}), and Counter Rudder Time Constant (T_{CR}). K_{CR} is a variable potentiometer calibrated from 1 to 10. T_{CR} is a preset switch of 6 settings from 3.5 to 28 secs.

(c) Permanent Helm (T_{PH}) is adjusted to apply sufficient permanent rudder to offset the drift caused by wind. The time constants available are from 100, 200, 400 and ∞ seconds.

(d) Damping (T_D). This imposes the time delay on rudder demand to allow for recovery action to take place within the natural yawing motion so that the steering gear is not subjected to repeated cycles of activity unnecessarily. The range provided is from 0 to 3.0 seconds.

Sea wave simulation was obtained by modelling around a typical wave height spectra obtained from the North Atlantic. By choosing a filter with a transfer function resembling the shape of the wave height spectrum and using white noise to excite the filter a fairly realistic simulation can be achieved. Analogue filters were used to prevent interface problems with the rest of the equipment. The circuit consists of a pseudo random binary sequence (P.R.B.S.) generator and a butterworth filter. By varying the amplitude of the PRBS controllable wave height is obtained. The system is shown below:-

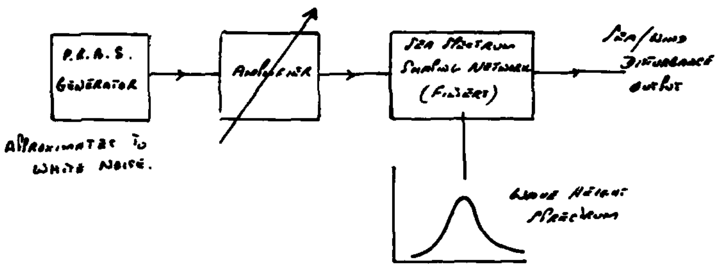


Figure 8. Sea/Wind Disturbance Simulation.

2. ANALOGUE CLOSED LOOP SIMULATOR AND DIGITAL CHECK ON MATHEMATICAL MODELS

Analogue simulation was carried out by connecting the simulators as shown below. The output was recorded on an X-t recorder.

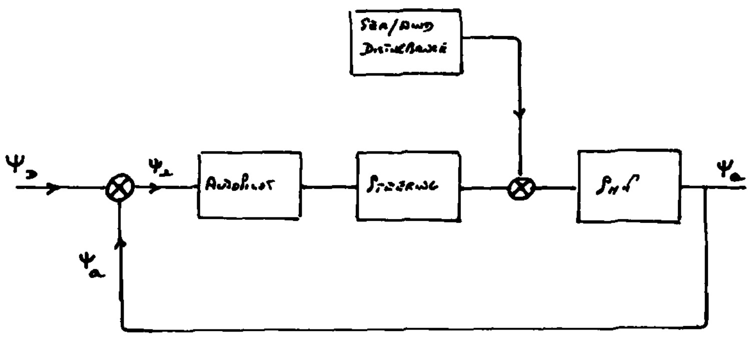


Figure 9. Closed Loop Connection of Simulator.

A number of problems arose with the device, the first being the fact that disturbance signals were fed back into the rudder input via the feed back line in the absence of a buffer stage. Secondly the recording equipment was inadequate at that time, and only one reading (yaw or rudder angle) could be recorded. Thirdly, the ship model was unsatisfactory for unstable ships.

Software Digital Simulation, using a Hewlett-Packard 9820A desk computer, was used with a graph plotter, and the accuracy of the transfer functions used for the analogue mathematical models checked by computer programme to ensure their accuracy. This was carried out by using a step response and comparing the analogue and digital results together. To obtain a software computer programme the transfer function was converted back to the time domain.

The step response of an input $\frac{1}{S}$ to an equation, $\frac{K(1 + ST_3)}{S(1 + ST_1)(1 + ST_2)}$ will be

$$V_0(S) = \frac{K(1 + ST_3)}{S^2(1 + T_1S)(1 + T_2S)}$$

By partial fractions, this can be shown to be:-

$$K\left(\frac{A}{S} + \frac{B}{S^2} + \frac{C}{(1 + T_1S)} + \frac{D}{(1 + T_2S)}\right)$$

and through inverse transforms,

$$V_0(S) = K\left(A + Bt + \frac{C}{T_1} \cdot e^{-\frac{t}{T_1}} + \frac{D}{T_2} \cdot e^{-\frac{t}{T_2}}\right)$$

$$\text{where } A = \frac{1 + T_3}{(1 + T_1)(1 + T_2)} - 1 - \frac{T_1^{\,2}(T_1 - T_3)}{(T_1 - T_2)(1 + T_1)} - \frac{T_2^{\,2}(T_2 - T_3)}{(T_2 - T_1)(1 + T_2)}$$

$$B = 1$$

$$C = T_1^{\,2} \cdot \frac{(T_1 - T_3)}{(T_1 - T_2)}$$

$$D = T_2^{\,2} \frac{(T_2 - T_3)}{(T_2 - T_1)}$$

By substituting the various values of K, T_1, T_2, and T_3 the response of the model can be found. The results were compared with the analogue graph and were found to be identical.

The unstable ship model could now be simulated by the same technique using the flow diagram in figure 10.

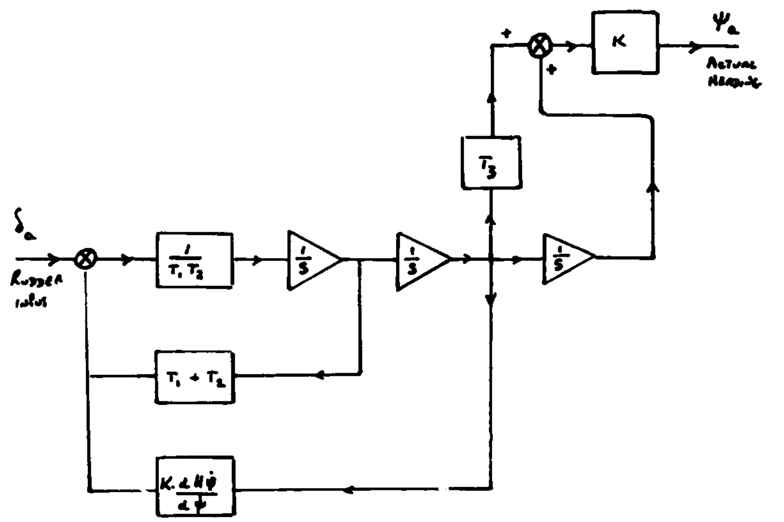


Figure 10. Flow Diagram of Unstable Ship.

A step response and 20° zig-zag test was carried out. (A modified z test was simulated which showed that if steering was ordered before the rate of change was fully developed a less stable ship can be steered like a stable vessel.)

The autopilot model was similarly checked by computer programme, together with the steering system and the sea/wind disturbance device. The software programme was based on the flow diagram of figure 11 below.

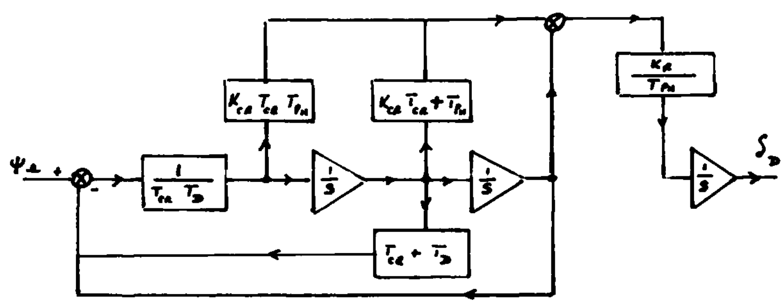


Figure 11. Flow diagram for digital simulation of autopilot.

The options open for both analogue and digital simulation are extensive, but a useful study was thought to be a comparison between Bech's (4) and Koyama's (7) autopilot design settings to reduce propulsion losses.

Bech (4) has advocated a flat wide closed-loop frequency response with high steering activity to reduce hull losses whilst Koyama (7) has emphasized that even if yawing is considerable the increase in hull resistance is minimum and recommends a low bandwidth controller with small rudder losses. For the three vessels, Mariner Class, Atlantic Song and Sea Splendour, various values of K_R, K_{CR}, and T_{CR} were tri d in order to find the right settings to fit the two criteria and it was concluded that under Bech's criterion the settings would be:

		K_R	K_{CR}	T_{CR}
A	Mariner Class Vessel	3	9	5
B	M.V. Atlantic Song	1	9	5
C	M.T. Sea Splendour	3	9	11

This was established by computer programme to provide the frequency response. (T_D and T_{PH} were ignored.)

For Koyama's design all three models need to take the lower limit of the settings, i.e. $K_R = 0.5$ $K_{CR} = 2$ $T_{CR} = 28$.

The Bode plote for Ship A is given in figure 12 and it can be seen from there that the overall bandwidth is directly proportional to the rudder output - i.e. a wide bandwidth also has a high rudder amplitude. Thus over a sea spectrum in a range of say 0.4 to 1.2 rads/sec the steering system will work hard (rudder response is approximately 40 dB's) under Bech's criterion whilst under Koyama's design the rudder has much less movement (response approximately 15 dB's) and the hull movement is allowed to continue.

3. ANALOGUE AND DIGITAL SIMULATOR RESULTS

Figure 13 shows the course keeping qualities of the three linear ship models under the two design criterion based on analogue simulation.

Figure 14 shows the high rudder activity to maintain this course under Bech's criteria for Ship B only. (Similar results occurred for Ships A and C.)

Figure 15 shows the effect of increasing the dead band width on the rudder activity.

Figure 16 shows the effect of a 10° course change on Ship A. Under Bech's criteria a 40° rudder swing occurred to produce the change whilst only 12° was required in Koyama's design although the response time was much quicker by Bech.

Figure 17 shows the course keeping performance of a non-linear ship when disturbed. The rudder worked hard for Bech but the ship maintained its heading although offset. For Koyama, rudder activity was small and effected by the steering system limit cycle and did much less work. The head drifted off however in spite of the gradual increase in rudder offset angle but judging by the trend it will probably restore the condition when the rudder angle is as large as Bech's in the early stages.

Figure 18 shows the effect of increasing the dead band width on course keeping quality. The narrow deadband of 0.5 in figure 17 shows in the rudder limit cycle and this disappears when the deadband is increased. It returns sporadically in the Koyama model beyond 2.5° and resettles.

Finally, figure 19 shows the course changing performance of a non-linear ship which is similar to the linear vessel. This was because such a large demand steered the vessel into the stable region.

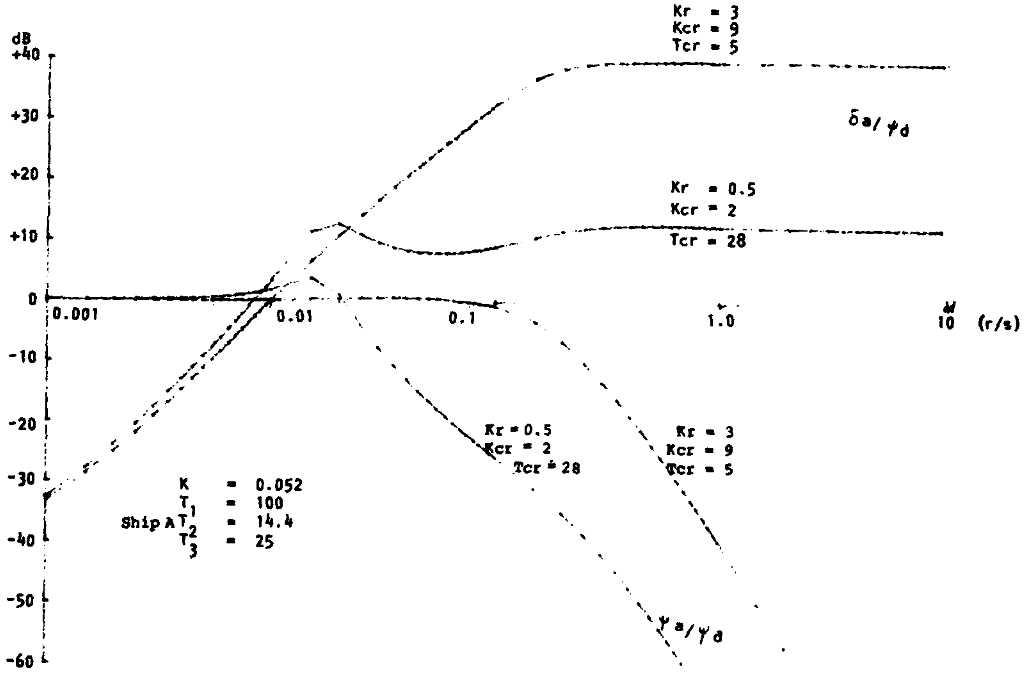


Figure 12. Bode Plots Relating Actual Course to Course Demand (ψa/ψd) and Rudder Output to Course Demand (δa/ψd).

	Ship Parameters			
	K	T_1	T_2	T_3
A	0.05	100	13.8	23.2
B	0.11	168	20	114
C	0.04	147	20	29.4

Steering System

$$\frac{\delta a}{\delta d}(s) = \frac{1}{(1 + TrS)} \qquad Tr = 2.2$$

Autopilot

$$\frac{\delta d}{\Psi e}(s) = \frac{Kr\ (1 + Kcr\ Tcr\ S)}{(1 + Tcr\ S)}$$

Td = 0

Tph = ∞

Disturbance KD = 5

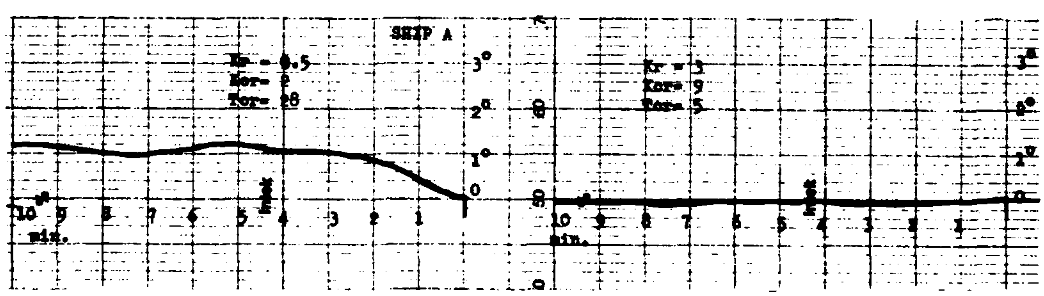
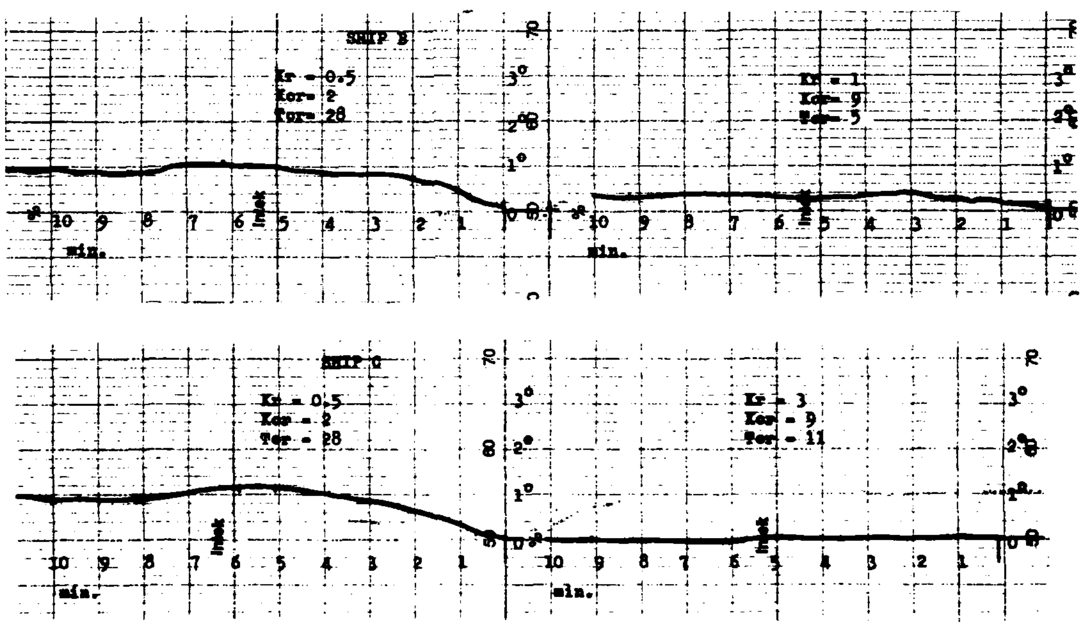


Figure 13. Closed-Loop Analogue Simulation of Autopilot/Ship System in Course-Keeping Mode



/Cont'd. Figure 13. Closed-Loop Analogue Simulation of Autopilot-Ship System in Course Keeping Mode.

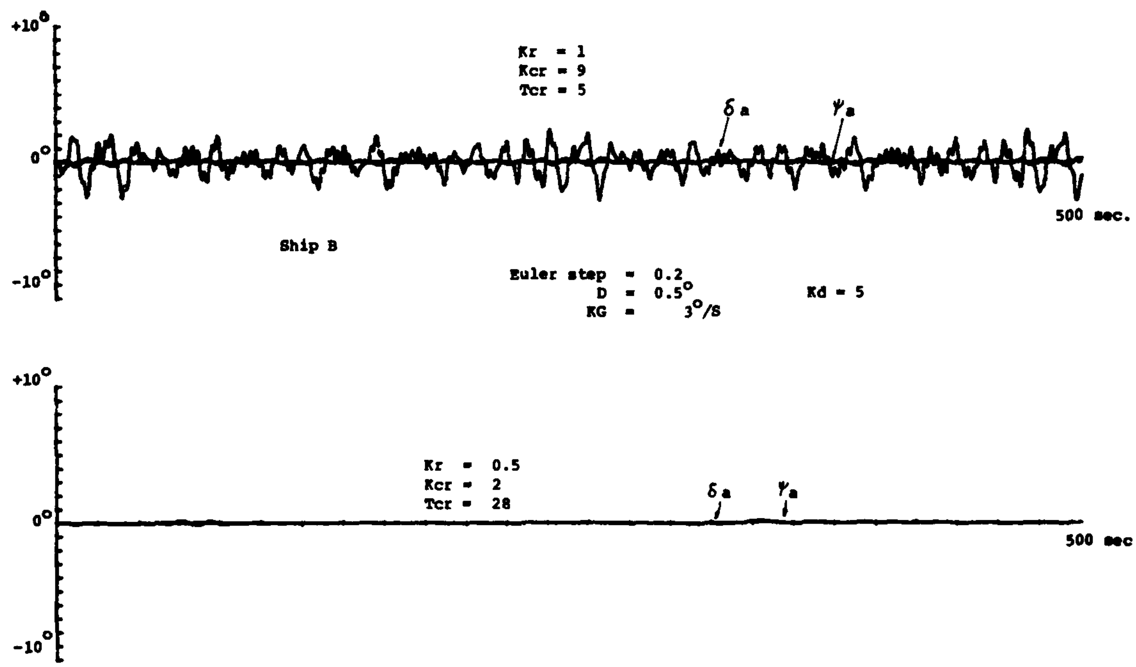

+10°
Kr = 1
Kcr = 9
Tcr = 5
δa
ψa
0°
500 sec.
Ship B
-10°
Euler step = 0.2
D = 0.5°
KG = 3°/S
Kd = 5
+10°
Kr = 0.5
Kcr = 2
Tcr = 28
δa
ψa
0°
500 sec
-10°

Figure 14.

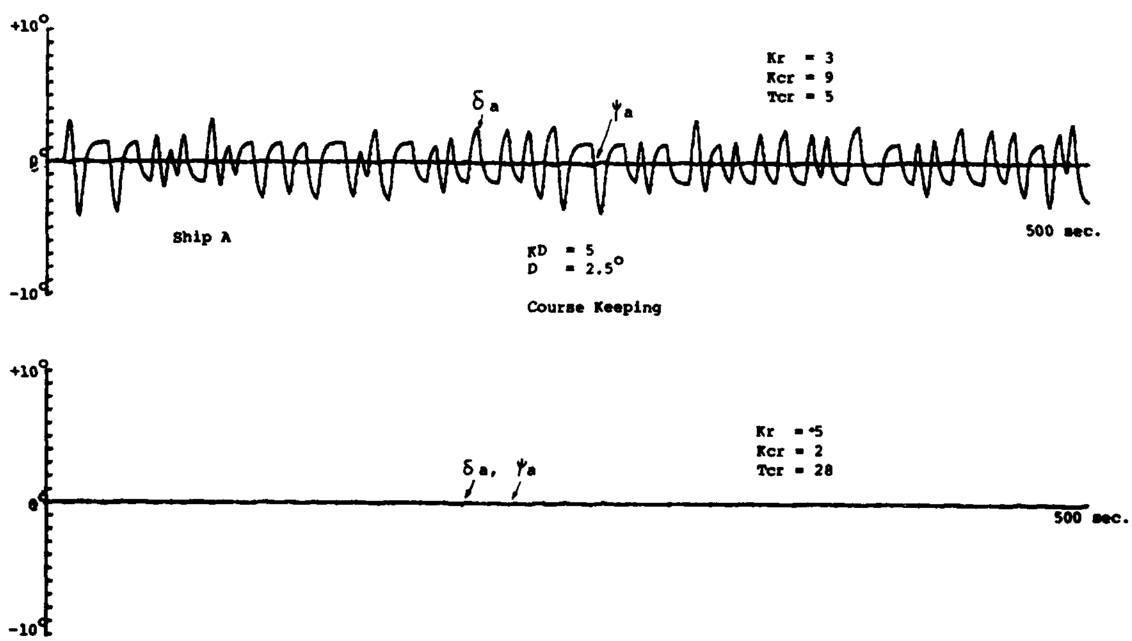


Figure 15. Effect of Increasing Deadband on Course-Keeping.

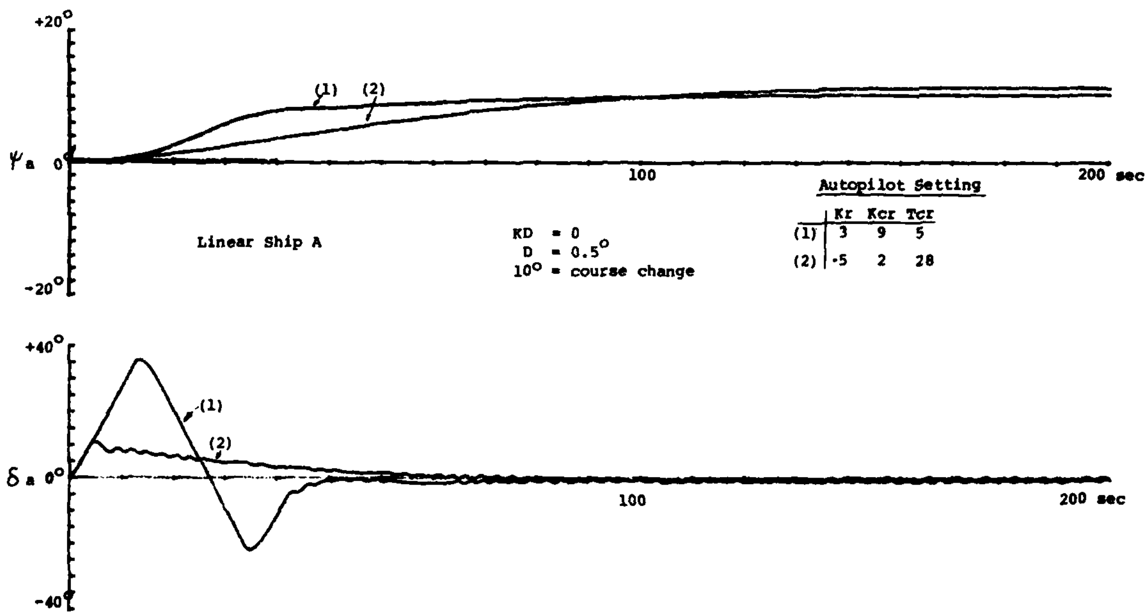


Figure 16. Simulation of Course-Changing by Linear Ship.

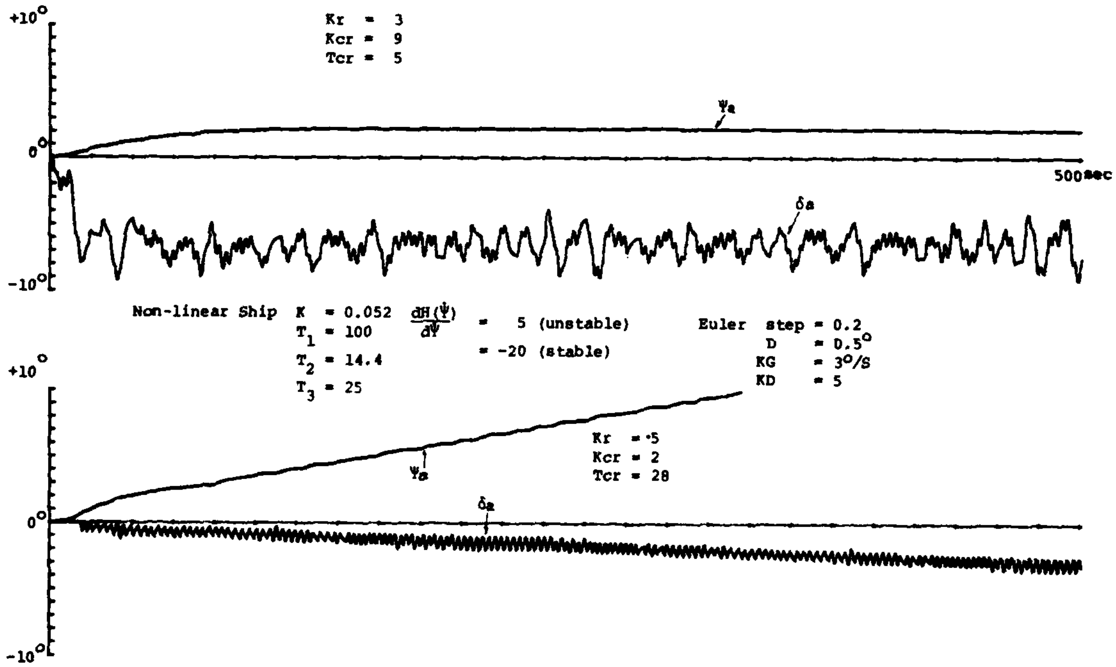


Figure 17. Course-Keeping Performance of Non-Linear Ship

Q 4-24

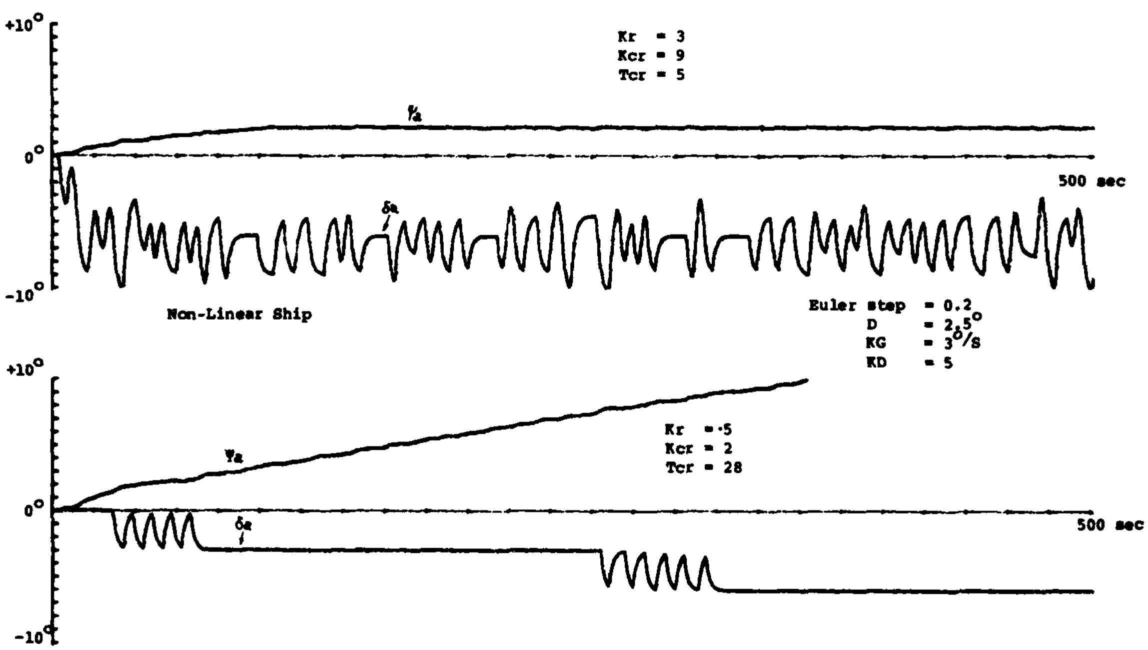


Figure 18. Course Keeping Performance of Non-Linear Ship with Increase Dead band

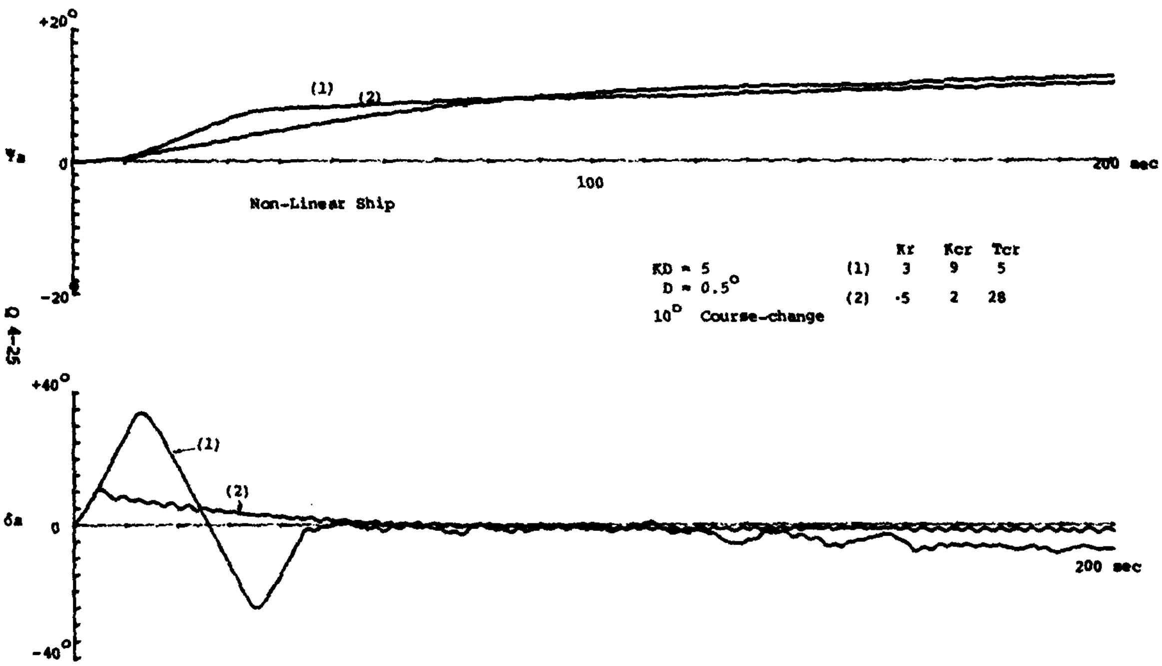


Figure 19. Course Changing Performance of Non-Linear Ship.

4. REFERENCES

(1) M. A. Abkowitz, "Lectures on Ship Hydrodynamics - steering and manoeuvrability," Hya Report, May 1964, pp.1-34.

(2) J. P. Comstock, "Principles of Naval Architecture," S.N.A.M.E., 1967.

(3) K. Nomoto et al, "On the Steering Qualities of Ships," International Ship Building Progress, 1957, pp.354-370.

(4) M. I. Bech, "Some Aspects of the Stability of Automatic Course Control of Ships," Paper 17, Vol. 14, 1972, J.M.E.S., pp.123-131.

(5) K. Nomoto, "Problems and Requirements of Directional Stability and Control of Surface Ships," Paper 1, Vol. 14, No. 7, 1972, J.M.E.S., pp.1-5.

(6) K. J. Astrom and C. G. Kallstrum, "Identification of Ship Steering Dynamics," Vol. 12, Automatica, pp.9-22.

(7) T. Koyama and S. Motora, "Some Aspects of Automatic Steering of Ships," Japan Ship Building and Marine Engineering, Vol. 3, No. 4, July 1968.

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